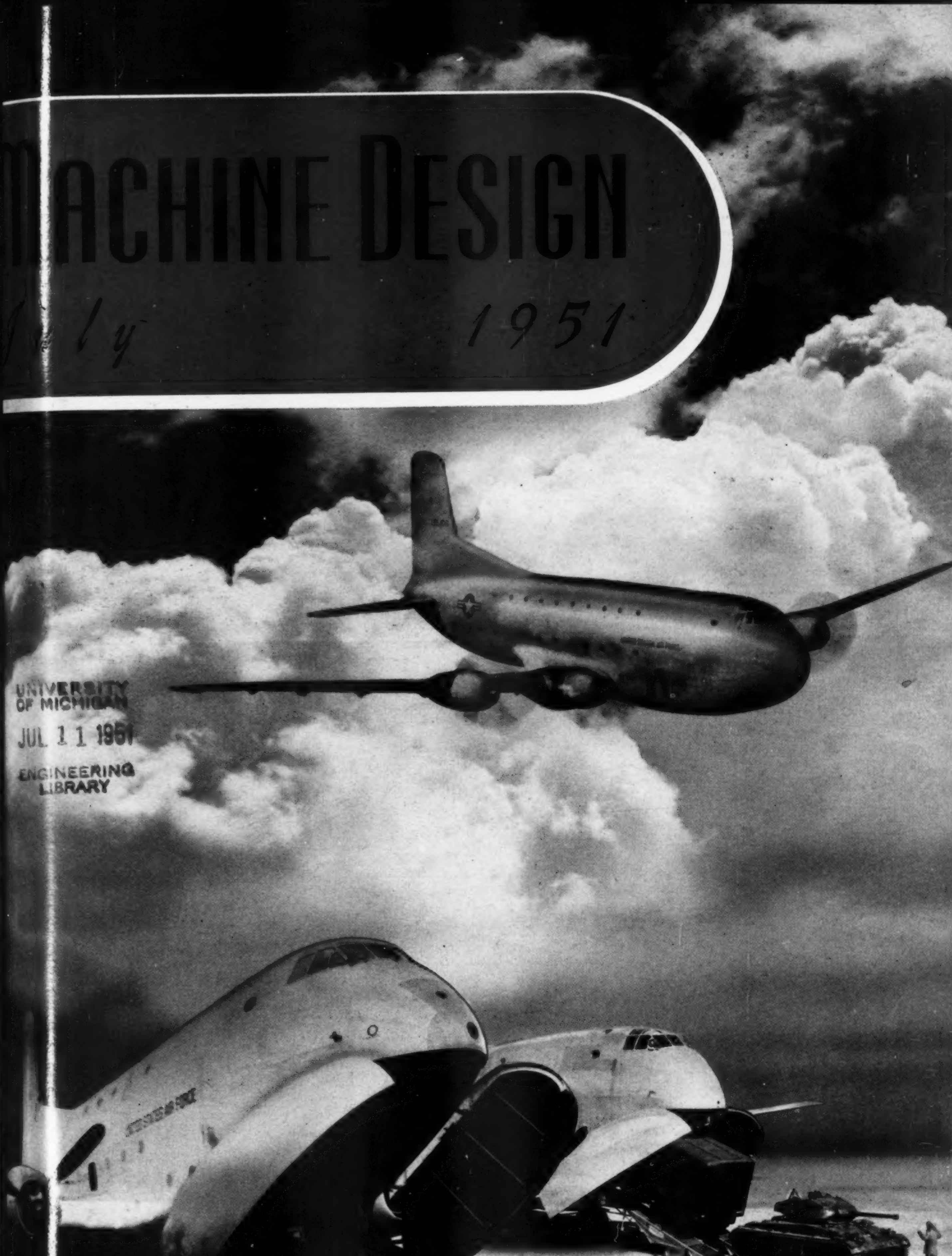
MACHINE DESIGN

July 1951



Aircraft Loading Ramps

(Contents, Page 3)

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Published on the seventh of each month. Subscription in the United States and possessions, Canada, Cuba, Mexico, Central and South America: One year $10. Single copies, $1.00. Other countries one year, $15. Copyright 1951 by The Penton Publishing Company. Acceptance under Act of June 5, 1934. Authorized July 20, 1934.

MACHINE DESIGN

THE PROFESSIONAL JOURNAL FOR ENGINEERS AND DESIGNERS

Vol. 23—No. 7 **July 1951**

DESIGN FOR PRODUCTION • STYLING • MATERIALS SPECIFICATION • DESIGN ANALYSIS • MACHINE COMPONENTS • ENGINEERING MANAGEMENT

Over the Board

Introducing a New Editor

To the roster of MACHINE DESIGN editors has been added the name of Robert L. Stedfeld, replacing that of Robert E. Denega. The new Bob, who joins us as Assistant Editor, is a graduate of Western Reserve University. He also studied engineering courses at Columbia and M.I.T. During World War II he was a meteorologist and operations officer with the rank of lieutenant in the U. S. Army Air Forces. Among his first assignments on MACHINE DESIGN is the preparation of New Parts and Materials announcements, the Engineer's Library, Topics, and the Itemized Index. In his spare time Bob likes to bowl, play the piano, and do card tricks.

This Month's Cover

That neat, round hole in the nose of the Douglas C-124A cargo plane shown on this month's cover isn't the result of enemy action or negligence on the part of the pilot. It's the air intake for a novel loading-door sealing system which uses ram-air pressure in flight to inflate the seal. In the article starting on Page 100 George Ikola, chief of structures at Douglas' Long Beach division, tells about this and about some of the trials and tribulations of fitting a 50,000-pound capacity loading ramp into the restricted confines of an airplane. The cover illustration, incidentally, is a composite job planned by "Mac" MacBain of our Art Department from four separate shots—a permissible exercise of artistic license, we think. Even though you may be looking at three views of the same plane, we hope you like the effect.

Wave Springs

How MACHINE DESIGN articles sometimes originate from readers' suggestions is well illustrated by the one beginning on Page 109. A machine designer asked us where he could obtain information on designing wave springs. Believing such information was generally available, we said we'd find out. Our searches disclosed nothing of consequence, so we consulted some spring authorities who steered us to the author of the article you see in this issue. We think the subject is presented in an eminently practical fashion and hope you will find it useful. Perhaps a wave spring is just what you have needed for a long time.

Materials Problems

On this page last month we told you that your replies to our questionnaires showed materials selection to be your second most pressing problem, topped only by that of reducing cost. The latest returns show a slight change in emphasis, the problems of production methods and improved appearance having returned to their original positions as second and third most important, leaving materials selection in fourth place.

Does this change mean that engineering materials problems are being successfully licked? Or does it mean that CMP has taken a great load off designers' minds? We doubt it, and we're still looking for the answer. Any ideas?

Quicker Than the Eye

Not even sound can match an artillery shell or rifle bullet when it comes to speed. Muzzle velocities of two and three times the speed of sound are the order of the day. A device which can, in a sense, "outrun" the projectile is the famed ENIAC at Aberdeen Proving Ground. On a visit there with a group of other SBME members, we saw ENIAC in action and were told that the computer could calculate the trajectory quicker than the bullet could travel it.

In case you don't remember, the letters in ENIAC stand for Electronic Numerical Integrator and Calculator. One of the early so-called Giant Brains, ENIAC went to work in 1946 and has since continued to amaze engineers and laymen alike with feats such as this.

Plastics for Big Stuff

We usually think of plastics as associated with small machines. At the same time, marine machinery is definitely in the large class. Predictions that "never the twain shall meet" have been overthrown by the appearance of the plastic gear housing discussed by Stark and Sawyer in the article beginning on Page 122. What makes it even more odd is that the plastic is teamed up with glass, of all things, to produce a material that will withstand the rough and ready treatment of a ship's engine room. Accustomed to heavy cast iron and steel surfaces, the crew have been known to walk and jump and even drop heavy wrenches on the tops of gear cases. The successful performance of the plastic housing shows what engineering plus imagination can do in meeting stringent requirements.

ITEMIZED INDEX

Classified for convenience when studying specific design problems

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Production

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TOPICS

POWERFUL PEA-SIZED MAGNETS have a lifting power 24 times greater than Alnico-5. Although these permanent magnets are described as the most powerful in the world in small sizes, extensive commercial use is not foreseen by A. H. Geisler and D. L. Martin, General Electric scientists who developed the magnet. Reason is the prohibitive expense of the large amounts of platinum used in the alloy.

ATOMIC BY-PRODUCTS for industry are being studied in a survey by the Stanford Research Institute. Vast quantities of radioactive by-product material are produced by the government's atomic energy plant at Hanford, Wash. Almost all these radioactive materials are unused and being stored in vaults—due to lack of practical applications. Part of the program, a prospectus suggesting possible industrial uses, has already been completed. The second phase, to determine specific industrial processes and products in which fission products can be used, is now under way.

POWDER-FED FLAME burns and blows away excess surface metal and sand encrustations on castings. Iron-rich powder is fed through an oxy-acetylene flame to a low-velocity oxygen stream, producing superheated liquid iron oxide. When the powder-fed flame is directed against a casting, the metal surface is brought up to kindling temperature, oxidized and blown away. The new "powder washing" process was originated by Linde Air Products.

SAND—or, rather, a silica material made from sand—has been proposed for thickening oil in making grease. The new silica, still in the experimental stage at Du Pont, comes in the form of balls less than a millionth of an inch in diameter and is treated to float on water. The silica material helps the grease resist water, mechanical breakdown and heat.

ROBOT WEATHER STATION releases its parachute after being dropped from an airplane, stands up by itself, raises a vertical radio antenna and starts sending weather reports. Called the "Grasshopper" and developed by the National Bureau of Standards for the Navy, the device uses small explosive charges to disengage the parachute, extend the 20-foot telescopic antenna and release six spring-operated legs. A timing mechanism starts automatic radio transmission of temperature, pressure and humidity reports.

HARD COATING for magnesium which can scratch glass or polish steel of Rockwell 65 C hardness has been developed by the Army. The coating is applied electrolytically in films as thin as 0.0015-inch. Heat-tested magnesium panels have melted and flowed away from the hard coating, which remained undamaged.

SILICONE RUBBER can be bonded to glass, ceramics, aluminum, steel, etc., by a new process developed by General Electric. The bond is stronger than the rubber itself. Requiring a liquid primer, the process is adaptable to a variety of rubber-metal molded products and rubber-glass laminated structures.

PORCELAINED V-RINGS on commutators may replace presently used mica if experiments at Allis-Chalmers prove conclusive. Porcelain is claimed to have more uniform mechanical characteristics and greater gripping action than mica.

DENTAL-WAX WAFERS increase cutting life of punches and dies at the Northern Electric Co. Ltd. in Toronto. The wax wafer is pressed edgewise against the cutting edge of a worn tool, and a 50:1 enlargement of the resulting profile is studied in an optical comparator. Amount of dulling is thus measured, and no excess metal is removed when regrinding the tool.

SCRAMBLE FOR ENGINEERS will get worse in the next several years, according to John K. Hodnette, vice president of Westinghouse. Engineering graduates this year will total only 32,000, and in 1954 the estimated total will go down to 17,000. This situation, along with possible armed forces requirements, will create a serious shortage.

ATOMIC RADIATION DETECTOR of pocket size has been developed by the Navy. Weighing two pounds and powered by two flashlight batteries, the new Radiac will probably replace a Geiger instrument which weighs 24 pounds.

JULY 1951

MACHINE DESIGN
A PENTON PUBLICATION

A Good Place To Work

IN THE career of any man there comes a time when his salary levels off. The job he is doing is worth so much money and will be done no better by a man with ten years experience than by one with only five. Unless or until a promotion comes along he has reached a plateau.

Chances are that when the realization dawns, the average man suffers considerable discouragement. He lies awake at night, worrying about where he is or is not getting. If he is thoroughly matured he will take stock of himself and plan according to his talents. Perhaps promotion must await the passing of someone higher up, in which case he will contain his impatience and strive to qualify himself for the higher job. Perhaps the odds are strongly against a future promotion. Then there are two alternatives: accept the situation or look for another job.

At this critical stage management can influence the man's decision favorably by making him feel that in spite of his salary level his company is still a good place to work. One of our good readers, who is chief engineer for a well-known machinery manufacturer, tells us of a simple plan which is working well in a number of engineering departments. Here it is in his own words—a constructive approach to an ever-present problem facing engineering management:

Without too much fanfare, the company quietly recognizes a preferred group of employees. The group includes certain key people but will also include engineers and designers in nonexecutive positions who have done and are doing an excellent job but who have reached a salary plateau through the simple fact that there is a ceiling price tag on their jobs.

Membership in the group carries certain privileges. The engineer is urged to join a technical society to which he is eligible and which parallels his work. The company pays his annual dues and assures him of one week's attendance at any national meeting at company expense. The week can be split over more than one meeting if desired. Alternatively the engineer may elect to attend national trade shows and expositions relating to his activities on the job.

Other privileges can be extended, such as the day off on the first day of the hunting season for a hunting enthusiast. In any event, none of these activities count against regular vacation time, nor is the time they take granted as a favor. In other words, the engineer has a right to expect the privileges by virtue of his membership in the preferred group.

What does it cost and what does the company obtain from all this? For each member of this preferred group the company pays perhaps $10 to $25 in society dues plus a week's travel expenses, say $75 in addition to train fare. For this money, which could easily be spent in over-entertaining customers or in unnecessary trips by a salesman, the company gets a better satisfied employee, one who has every chance of developing in caliber and breadth of view as a result of broader contacts. It is aimed at keeping the "salary plateau" man content that he has a *good* job. It is especially appropriate for engineering departments that abound in jobs handcuffing their men to desks and drafting boards.

Colin Carmichael

EDITOR

DESIGN OF
Aircraft Loading Ramps

By George H. Ikola
Chief of Structures
Long Beach Div., Douglas Aircraft Co.
Long Beach, Calif.

ABILITY to carry vehicles weighing 50,000 pounds without disassembly was a major design requirement of the C-124A airplane, *Fig.* 1. Cargo is loaded through the nose opening, vehicles entering and leaving the plane under their own power over two self-contained ramps. When in use, the ramps are inclined 17 degrees to the ground and in the same plane as the first section of the cargo compartment floor. Ramps fold to a vertical position in the forward section of the airplane during flight, with two large clam-shell type doors closing the nose opening. Design of the ramps was governed by the types of vehicles to be carried. The plane is capable of handling vehicles from jeeps to gasoline truck-trailer combinations and many caterpillar track-type pieces of equipment, including some light-medium tanks. Ramps are three feet wide, approximately 28 feet long, and are adjustable laterally to accommodate varying tread widths. The ends are supported at the forward floor level by hinges and at the ground by a support attached to the forward portion of the center ramp section, as shown in *Fig.* 1.

A third support at the center of the ramp was considered but was discarded for a number of reasons: space for such a support was difficult to find when the ramps were retracted and stowed; adjustment for height of the ramp hinge from the ground would have required hydraulic jack or ratchet mechanisms which would have added an extra operation to set ramps prior to use; and overall ramp weight would have been increased.

Ramps are made in three parts each, section lengths being a function of the stowage space available in the airplane. A toe section 63 inches long folds onto the surface of the center ramp sections, the two sections being locked together manually by an over-center clamp. Center and rear sections are each approximately 135 inches long and are attached to each other by hinges at the top and restrained at the lower section of the joint by truss rods. The truss rods slide through pivoted trunnions attached to the ramps, *Fig.* 2.

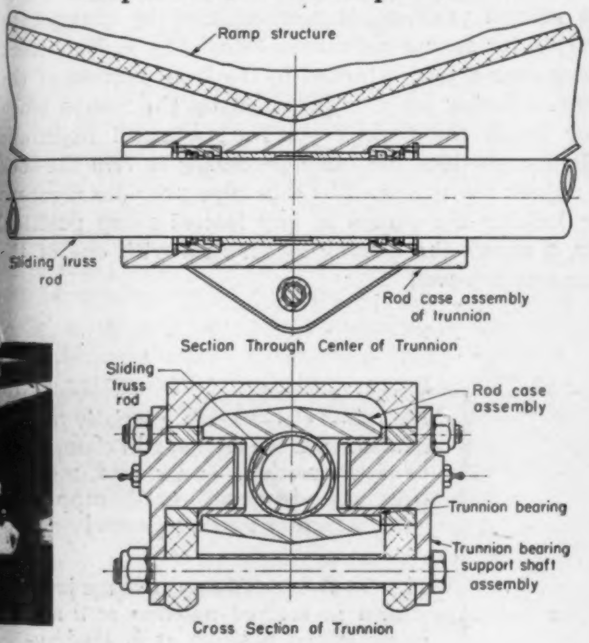
Center and aft sections are similar in construction. Main structural members on each side are of shear web and stiffener design, with cap members forming a triangular-shaped beam. Material used is 75 ST aluminum alloy extrusion for cap members and stiffeners and 75 STAL sheet for the web, with pivot end connections reinforced with 75 ST plates.

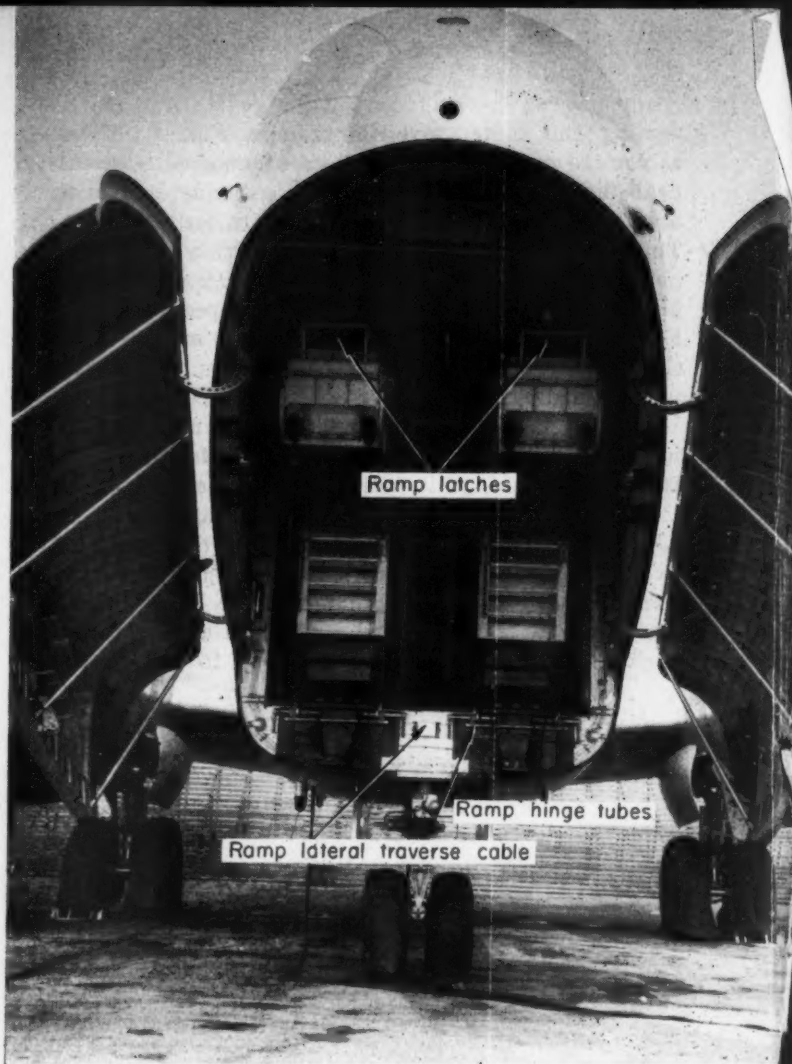
Tread surfaces are supported by ZK60A magnesium alloy extrusions attached to the side beams and spaced approximately 8 inches on centers, *Fig.* 3. A thin sheet of 75 STAL attaches to the top of the support structure and on top of this sheet, 0.460-inch plywood is bolted

Fig. 1—Left—Loading doors of C-124A cargo plane in open position, with ramp lowered. Toe, center and aft ramp sections can be seen, with truss rod system visible under center and aft sections

Fig. 2—Below—Drawings showing trunnion construction. Truss rods slide through pivoted trunnions as ramp is lowered or raised

Fig. 3—Right—Ramps in stowed position showing construction of ramp beams. Ramp wheels shown permit ramp to roll to full-down position





to the structure. Plywood is aircraft quality with double face plies of birch and five core plies of spruce. The double face plies are cross-banded, giving excellent properties in both directions. To develop a good tractive surface on the ramps many experiments were run with the 17-degree slope. Ramps were mocked-up prior to completion of the design to determine the action of vehicles on the slope. The first trial was with steel cleats measuring ½ by 1 by 10 inches bolted to the planking with various spacing. Tank treads pulled the cleats out and rubber on the tires of the wheeled vehicles was shaved when the wheels started to spin.

Cloth-Backed Ramp Covering Selected

A second trial was made using 2-inch washers, 3/16-inch thick, bolted to the planking, with the bolt heads projecting over the washer surface. This arrangement was no more successful than the first trial. The third trial was with a cloth-backed nonskid surface material consisting of coarse-grit particles cemented to the cloth backing which, in turn, is cemented to the plywood.

Tests with vehicles were made on this material under many conditions. Dry, wet, oily and muddy conditions were simulated. During the mud test, made with oil well drilling "mud" piled as thickly as possible, all vehicles were able to gain traction on the ramps. Several instances occurred where the vehicles failed to get onto the ramps because they could not get traction on the slippery cement pavement. These occurrences dramatically showed the superiority of the ramp surface. When traction is difficult to obtain on the slippery surface, a winch on some prime mover outside the airplane can be used to pull vehicles into the plane by cable.

For operating the ramps, a mechanism having three functions is required: (1) a transverse operation; (2) raising and lowering operations; and (3) a locking operation, with ramps stowed in the plane.

Transverse operation is handled by a cable system connected to a two-way hydraulic cylinder. The two ramps are synchronized to maintain equal tread distances to the center line of the airplane. Transverse adjustment is made with the ramps off the ground and not latched in the stowed position. Ramps slide on hard chromium-plated steel tubes that serve as the hinge attachment to the airplane, as shown in *Fig.* 3.

Raising and lowering are performed by a cable system actuated by a one-way hydraulic cylinder. Each of the ramps is attached to its hoist cable by a bridle connected to the ramp near the hinge con-

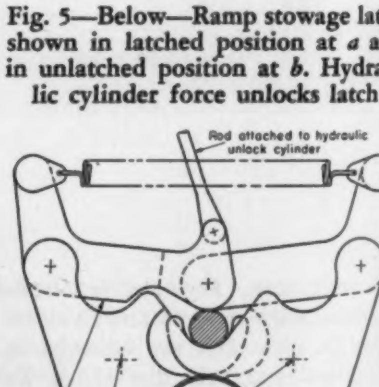
necting the center and rear ramps, *Fig*. 4. These cables are routed aft and up to a common attachment near the top of the airplane. Here a single cable is attached to the hydraulic piston that raises the ramps. This cylinder is 180 inches in length with bore of 1.5 inches and operates at 3000-psi pressure. Hydraulic pressure is required for the raising operation only. For lowering, weight of the ramps is sufficient to energize the system, the hydraulic cylinder furnishing a retarding force.

During the lowering operation, it is necessary to impart an extending force to the center ramp, to move it outward as it comes down rather than back toward the airplane. This motion is accomplished by a hydraulic cylinder which occupies the space of one of the truss members on each ramp, *Fig*. 4, and functions as such when the ramps are down. When the ramps touch the ground a small inflated wheel on each ramp, *Fig*. 4, allows the ramp to roll to the full-down position. When the ramps are in use, the load depresses the tires until the primary ramp support structure touches the ground. It is, therefore, not necessary to have a wheel capable of carrying the full ramp load. With ramps down, the front section is unlocked from the center ramp and manually rotated about its hinges to the open position. The ramps are now ready for use.

The third or locking operation, with the ramps in the stowed position, is accomplished by clamp-type over-center spring-restrained locks, *Fig*. 5. Each lock closes over a tube attached to the front section of the ramp. Closing force is provided by the ramps when they reach the stowed position; a small hydraulic cylinder provides the force necessary to trip the lock to unlock the ramps. The tube also provides a means for locking the ramps in any lateral ramp position. *Fig*. 5 shows the locking mechanism with one of the supports removed.

Fig. 4—Left—Ramps partially extended, with toe section still clamped to center section. Under load, ramp tires are depressed until support structure touches ground

Fig. 5—Below—Ramp stowage latch shown in latched position at *a* and in unlatched position at *b*. Hydraulic cylinder force unlocks latch



Three design changes were found to be necessary after the ramps were completed. During retraction of the ramps, the sliding truss system operated independently on each side of the ramps. Difference in friction on the tubes caused differences in the sliding rate of each truss, thereby introducing chatter during the folding operation. This condition was corrected by tying the truss rods to each other through a rigid cable system.

Cable Loads Cancelled

A second change was caused by twisting of the cable that is attached to the retracting piston. This twisting caused rotation of the piston as it went through the tube. The piston is now restrained from rotating by a square mandrel attached to the piston rod and guided by a square hollow tube.

The third change was caused by the load in the cable system when the ramps are in the stowed position. Cable routing is such that changes in cable direction are necessary to reach a common terminus. Pulleys over which the cables pass are mounted on a common bulkhead and project about 6 inches from the bulkhead face. Loads on these pulleys caused deflections in the bulkhead and pulley supports, which were transmitted to the primary structure of the airplane. These deflections were considered to be more than should be allowed for good service life. A direct compression tie was added between pulleys close to the pulley support bolts. The high loads in the cable system opposed each other so that the loads did not have to go through the surrounding structure.

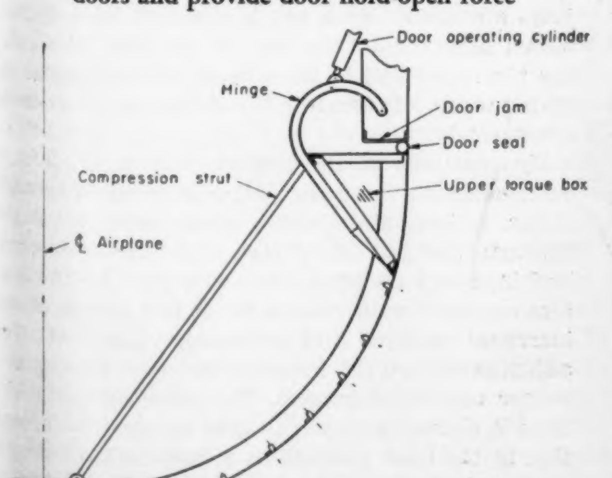
The nose doors are designed for use on ground operation only. They are shaped like two large clam shells, with each door approximately 22 feet long and 8 feet wide. The door structure consists of two torque boxes at the longitudinal edges, with conventional frame, skin and stringer construction between the torque boxes and compression struts connecting the torque boxes, *Fig.* 6. A torque box at the top outer edge of the door supports the door hinges and door-to-jamb latches. The torque box at the edge near the bottom center line of the airplane supports the center door latches and the compression struts provide rigidity to the door and assist in transmitting center-latch loads to the upper torque box, *Fig.* 6, to which the hinges are attached.

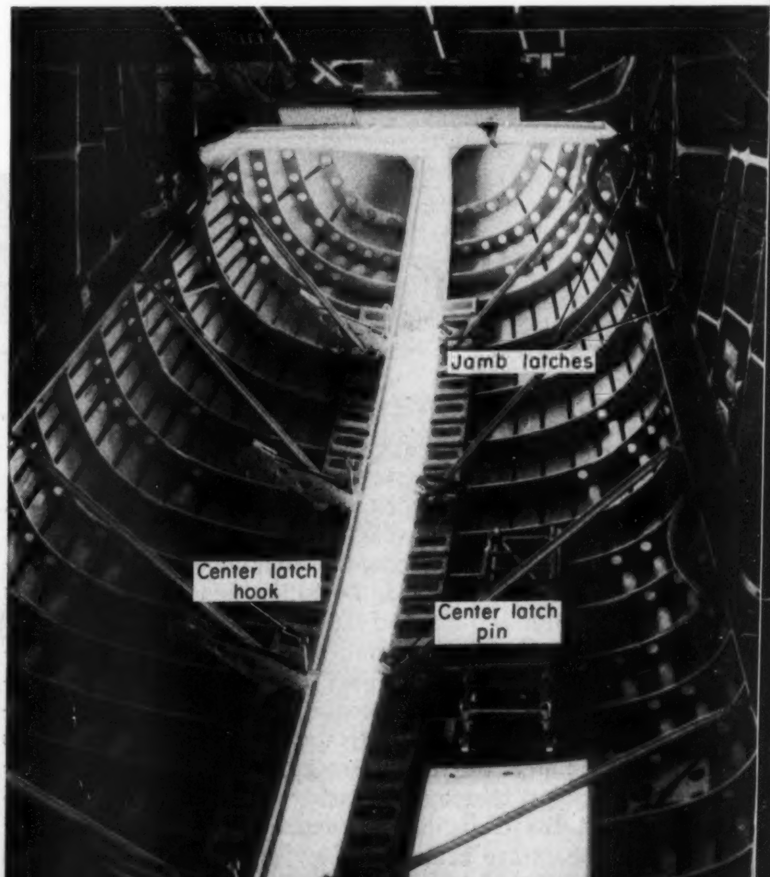
Material used for the door structure is mostly 24 STAL aluminum alloy. Some 75 ST is used at the torque boxes where load concentrations and shapes are such that the higher allowable tension and compression stresses for 75 ST give a weight saving. Struts are 24 STAL. They are attached to the steel end fittings by a roll-swage process developed by another manufacturer and are purchased complete. This swaging operation is sufficient to develop the full compression and tension value of the tube, thus eliminating a cumbersome tube end fitting design.

Each door is supported by two hook type hinges made from square steel tubes and are operated by hydraulic cylinders attached to each hinge, *Figs.* 6

Fig. 6—Below—Section through nose of loading door showing construction. Upper and lower torque boxes are tied together with compression strut

Fig. 7—Right—Partially opened doors seen from inside. Hydraulic cylinders attached to hinges operate doors and provide door hold-open force





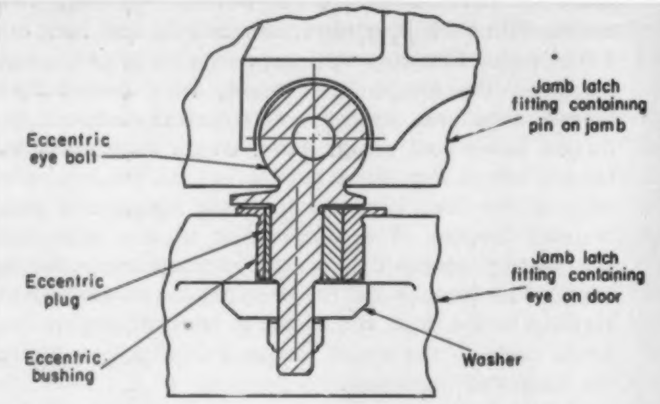


Fig. 8—Above—Latch adjustment mechanism. Rotation of eccentric plug and bushing gives 7/32-inch adjustment, eye bolt gives additional adjustment

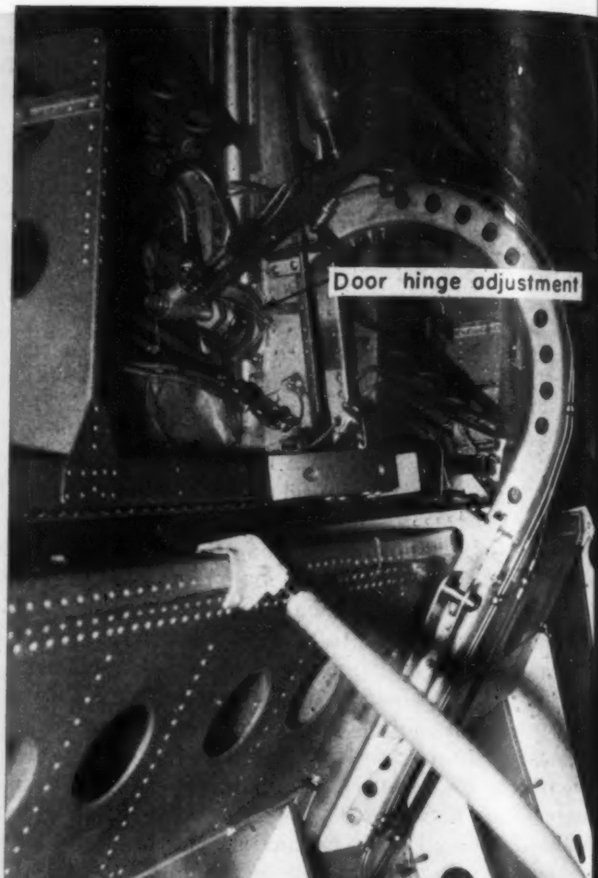


Fig. 9—Right—Close-up of door hinge. Door disconnect mechanism, *Fig. 10,* is located where hinge is attached to door

Fig. 10—Below—Door detaching mechanism. Normal pin position is shown at *a.* With pin control handle in emergency release position, *b,* pin has rotated from normal position to free door

and 7. These cylinders also provide the door hold-open force. The hinge pivot point is adjustable, allowing ⅜-inch adjustments fore and aft and ⅜-inch adjustment in the hinge plane about the nominal pivot point so that the plane of the hinges can be kept parallel. In the original design, none of these adjustments were provided. Deflections in the door hinges and adjacent structure caused difficulties when the doors were being installed. It was necessary to relocate the hinge support structure to obtain satisfactory door-to-jamb contour match and door operation. The addition of adjustable hinge support brackets eliminated the time required to relocate the hinge supports, thereby decreasing the time required to install the doors by approximately one third.

Doors are held closed by three latches along each door at the outboard jamb and by four latches along the airplane center line attaching the two doors together, *Fig.* 7. The three jamb latches are of the bayonet pin type with the eye portion of the latch attached to the door and the pin attached to the jamb structure. Each pin is operated by a hydraulic piston attached to the end of the pin. The axis of the pin is parallel to the side of the airplane so that with proper adjustment, the door will be restrained transversely.

Eye portions of the latches contain the latch adjustment, with vertical adjustment obtained by washer shims. Horizontal plane adjustment is obtained through the use of a series of eccentric serrated bushings and serrated eye bolts, *Fig.* 8. The eye is also eccentric with respect to its bolt center, allowing increased inboard and outboard adjustment. These adjustments aid in keeping the door and jamb in proper surface alignment. The slope on the point of the pin allows the latch to take up some slight variation in the door position as jointly influenced by the hinge cylinders and the latch itself. Forward and aft door loads are handled by one jamb latch on each side so that it is not necessary to obtain precise location of the other jamb latches in the fore and aft orientation.

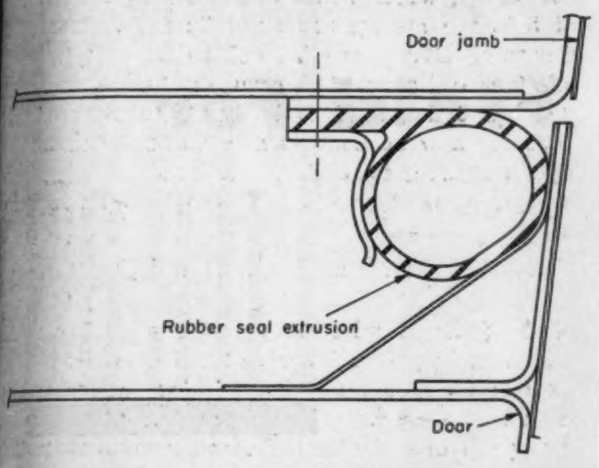


Fig. 11—Drawing showing nose door seal which is inflated, when in flight, by ram-air pressure from opening in nose of plane

Doors are latched to each other at the airplane centerline by means of four hydraulically operated latches, *Fig.* 7. Mechanical advantage in the hinge cylinder system may not be sufficient to overcome all the seal deflection forces that may result during severe cold weather operation. Therefore, the center latch has been designed with a hook that traverses a prescribed orbit to grasp and pull the doors together. Doors can be latched with gaps varying from zero to approximately four inches, the hook locking in the closed position automatically. When the hydraulic piston pushes to open the latch, the hook unlocks first and then is free to travel through its unlocking orbit. The hook and its mating bolt are 4130 CM steel, heat treated to 180,000-200,000 psi. A roller bearing attached to the hook rolls along the track surface to control the hook orbit.

Doors can be detached from the airplane if the hydraulic power system is inoperative. Center latches have a manual unlatch mechanism which releases the bolt that is grasped by the hook and jamb latches can be opened by a wrench which grasps each pin so that an axial force may be manually applied to unlatch. Hinges can be disconnected from the door at the torque box, *Fig.* 9, by manual rotation of a pin so that the hooks which grasp the pin are released, thereby disconnecting the door. *Fig.* 10 shows the emergency door release mechanism.

The item of next importance for proper functioning of the doors is proper seal design to prevent air and water leakage. A balance has to be maintained between seal wall thickness, Shore hardness of the rubber, and size of cross-section. It is necessary to keep the seal compression as low as possible to keep door closing force down. Size of the door jamb generally regulates size of seal so as to accommodate variations in structure and still obtain sufficient contact to maintain seals. *Fig.* 11 shows the seal finally selected.

Various types of seals were tested in a pressure test box. Seals with ridges in the longitudinal direction and smooth seals were tried. It was found that the ridged seal was not as effective as the smooth one. For water, especially, and air to some extent, the effect of ridges was to provide line contact for a distance and then, where an irregularity was present in seal or jamb, a leakage path occurred which allowed fluid to go along successive ridges until it found a path through the labyrinth.

Another factor in seals is the material available to produce the seal extrusion. The seal is required to function at a temperature range from minus 67 F to plus 160 F, to resist absorption of hydraulic fluid, to keep from checking when exposed to atmosphere and sunlight, and to withstand abrasion during door operation. Tests were conducted on numerous compounds furnished by a number of rubber suppliers since the YC-124A design and a compound closely approaching these requirements is being used.

Ram-Air Seal Pressure Utilized

For sealing purposes, the large cross section and relatively thin wall of the nose door seal required something besides its own rigidity or its own trapped air to keep its shape. A study was made of various means to sustain the shape. Use of an auxiliary air source to inflate the seals was believed desirable. Comparison was made between air pressure supplied by an air compressor and by in-flight ram-air from a plenum chamber near the nose of the airplane. The ram-air system is being used and effectively seals the doors. *Figs.* 1 and 2 show the ram-air intake in the nose of the plane.

All controls for ramp and door operation are located at a central control panel adjacent to the ramps. To assure proper sequence of operation, the various systems are regulated by electrical interlocks so that hydraulic control handles cannot be operated accidentally until the system is ready for the next phase of the cycle. Assuming doors are locked and ramp is in the stowed and locked position, the cycle is as follows:

1. To assure ground operation only, doors cannot open until landing gear oleo is compressed
2. Door latch handle is placed in unlatch position when doors are ready for opening. This starts the unlatch operation with the jamb latches opening first and then the center latches opening
3. Door opening handle is turned to *open* and door operating cylinders open doors to full-open position
4. Ramp release handle is turned to *lower* which causes the ramp lock to release and allows the ramps to lower. When ramps reach a predetermined position, the hydraulic cylinder operates to make certain that the ramps extend properly. Ramps extend to full down position and handle can be placed in *neutral*. The ramp toe can now be manually unfolded
5. When ramps are in the partially extended position (not touching the ground) it is possible to stop the lowering operation and operate the lateral traverse lever to increase or decrease tread width as required by the vehicle.

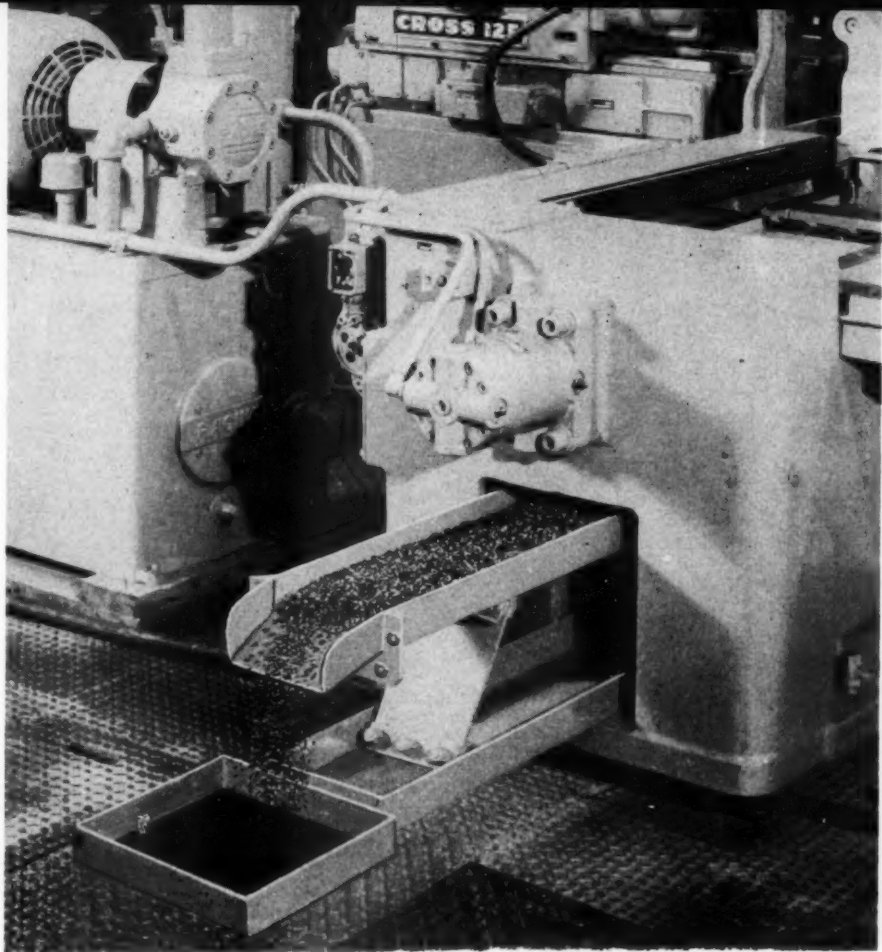
The reverse cycle is similarly controlled. It is possible to manually override the interlock if necessary, but care must be taken to complete each operation before going into the next step.

SCANNING the Field For Ideas

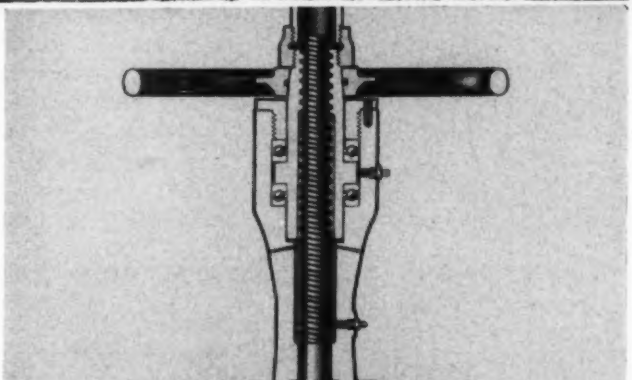
Automatic compensation for any small irregularities in the platen driving gears assures accuracy in the Swiss dividing engine, below. The correcting device, attached beneath the platen, provides an axial motion to the worm drive when required, thus supplying supplementary positive or negative displacement to the rotating platen. Designed by the Societe Genevoise d'Instruments de Physique and built for W. & L. E. Gurley, the machine graduates instrument scales accurately within one second error.

The platen has 720 teeth on its outside diameter and is operated by a single-thread worm, one revolution of the worm imparting an angular rotation of 30 minutes to the platen. The reference circle on the outer circumference of the platen is read with locating microscopes, the circle being graduated every 10 minutes of arc. Another feature that insures accuracy is the shock mounting of the machine. It rests on a block of concrete supported on coil springs that are placed, in turn, in a concrete foundation. The machine, in effect, floats and is thus independent of external vibrations. Glass covers provide protection from dust.

Oscillating conveyor, efficiently removing chips from the machines at top of next page, is mounted on one-piece flexible members which support the conveyor and function as springs in absorbing the energy of mov-



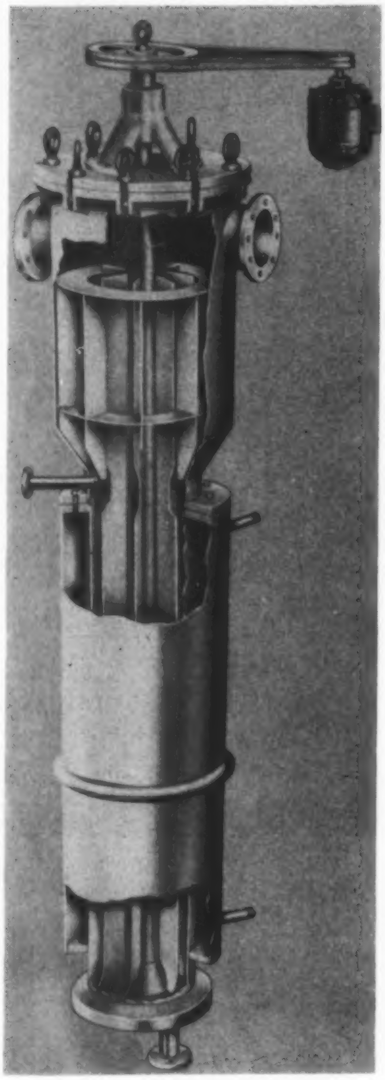
ment at each end of the conveyor stroke. In effect the conveyor oscillates at the natural frequency of the flexible supports, reducing the reaction forces on the drive to a minimum and requiring only the power necessary to convey material. The drive for this unit, designed by the Link-Belt Co., is a positive action, eccentric type with self aligning bearings, V-belt drive and flywheel.



Double-disk gate valve, below right, seats in both the open and closed positions. In the fully closed position, the disks wedge against both body seat rings. When in the full-open position, the disks again wedge against seat rings to seal off grease in the body from the line fluid and to form a fully enclosed path of flow called conduit flow. As shown in the illustration, only one half of the double disk is connected to the valve stem. The other half is held loosely to the first by four assembly links and by the action of spring-loaded plates floating on the valve-seat inserts.

Mating surfaces between the two disks include a double wedge—one angle assures tight seating when the valve is closed and the other, seating when open. At each end of travel, the floating half of the double disk comes in contact with stops to produce the wedging action. This wedge is released when the valve stem is turned in the opposite direction. The spring loaded plates press constantly against the double disk at its faces and hold the halves in contact while the disk is raised or lowered. In addition the disk is guided at its edges by long guide ribs in the body which are not shown in the illustration. As a protection against undue pressures in the valve body resulting from temperature change, a spring-loaded relief valve is provided in the bonnet of this unusual valve designed by the Crane Co. for use in the petroleum and gas industries.

A thin film surface produced in the rotary-vane evaporator at top of next page concentrates, in a single pass, many liquids such as food products which otherwise would be damaged due to the length of time required in passing through heated sections of other types of units. Designed by the Rodney Hunt Machine Co., the evaporator is a vertical tube with the lower portion surrounded by a heating jacket to heat the concentrating surface. The upper part of the tube, however, is not heated and is enlarged to

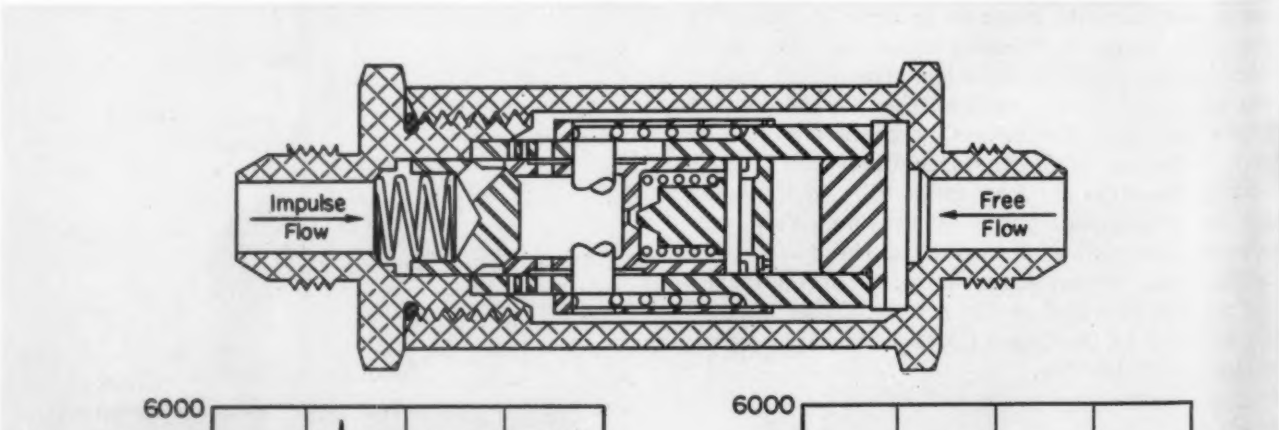


assist in "knocking-out" any droplets entrained in the vapors. Fins are fitted on the rotor and reach within 1/32-inch of the heated wall in the evaporator chamber to maintain a turbulent thin film.

In the upper zone of the unit, the centrifugal action of the vanes throws any entrained droplets to stationary separator fins which arrest their circular motion, permitting the material to fall back into the heating zone for evaporation. In this way liquids with a tendency to foam may be handled efficiently because the foam reaching the separator fins returns to the evaporator section. Material enters the unit at the left between the evaporator and separator, and the concentrate leaves from the bottom.

Surge valve, below, is designed to prevent the destructive shock set up in hydraulic power systems resulting from sudden starting or from sudden reversing flow in the system. The higher the pressure and the more rapid the shift of flow the more intense and destructive is the shock impulse, which may cause failure of tubing or fittings. This valve, designed by The Denison Engineering Co., is a normally-closed valve that opens when pressure is applied to the inlet side. It allows a gradually accelerated flow through the valve until fully open. After the period of acceleration, virtual free flow is maintained as long as fluid continues to flow. When the flow is interrupted, the valve resets and is ready for the next operation. It will open at a slower rate of speed when high pressures are in effect than at lower pressures. Also, the rate of opening will increase as the pressure in the outlet line approaches that at the inlet.

The valve is constructed with a spool which is normally held closed with a spring to prevent flow from inlet to outlet. However, when pressure is applied to the inlet, the spool moves toward open position. Rate of spool movement is controlled by a small pressure-responsive valve and by the difference between the outlet and inlet pressures. This results in the difference in opening speeds at various pressures. No more time is consumed in the operation of the valve than the duration of shock without the valve. Shown in the oscilloscope graphs below are typical durations for a 3000-psi system with and without the surge damping valve.



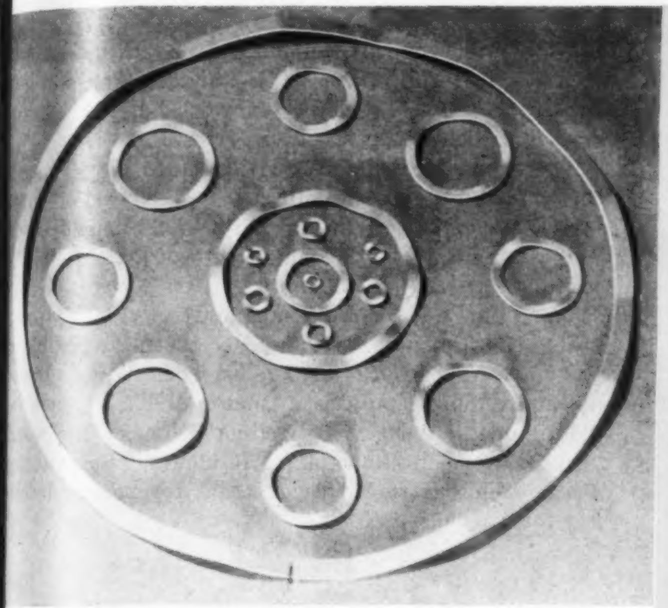
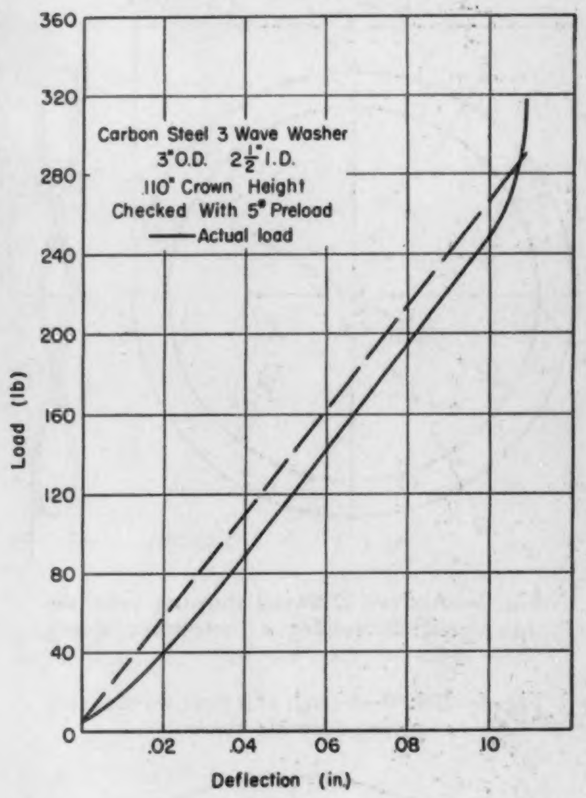


Fig. 1—Left—Groups of typical wave type spring washers showing size range

Fig. 2—Typical load calibration curve for a three-wave washer, below



Designing Wave Springs

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IN RECENT years the trend toward smaller and more compact machines and mechanisms has definitely decreased the amount of space available for dynamically and statically loaded parts. This is also true of the space allowed for the springs used.

When a static load or a small working range is required of a spring and the allowable amount of axial space is small, the use of a wave washer is an attractive method of obtaining the desired load. These springs are often used as cushion springs or cushion spacers between parts on shafts or to take up the expected amount of variation in the assembled parts.

Wave washers can be made in a rather large range of sizes, *Fig.* 1; however, when the ratio of the mean diameter to the radial width of the material becomes less than 8, the load and stress calculations are not accurate. This is due to the extreme distortion of the material and the fact that after deflection the waves tend to kink and are no longer uniform waves.

The uniformity of the height of the waves is of primary importance because the true load-deflection rate does not start until there is approximately the same load on each wave. For this reason a load-deflection rate is always checked with some preload. A typical calibration curve is shown in *Fig.* 2.

All edges and surfaces of wave washers should be free of burrs or rough surfaces as the friction caused by such edges and surfaces affects the load. The allowable solid stress for springs of this type made of carbon steel, hardened and tempered, is approximately 200,000 psi. In some special cases this figure is exceeded, but for general applications it should be used as a limit.

When designing wave washers, the allowable inside and outside diameter limits should be specified, because the wave washer will increase in diameter slightly upon compression, *Fig.* 3. This is a function of shape and material, and exact size should be left to the discretion of the spring manufacturer.

A washer can have any number of waves from three on up; however, in the normal range of sizes there are usually three, four or six waves. The formulas for calculating wave washers are as follows:

$$P = \frac{Efbt^3N^4}{1.94D^3}$$

$$S = \frac{12EftN^2}{\pi^2 D^2} = \frac{3\pi PD}{4bt^2N^2}$$

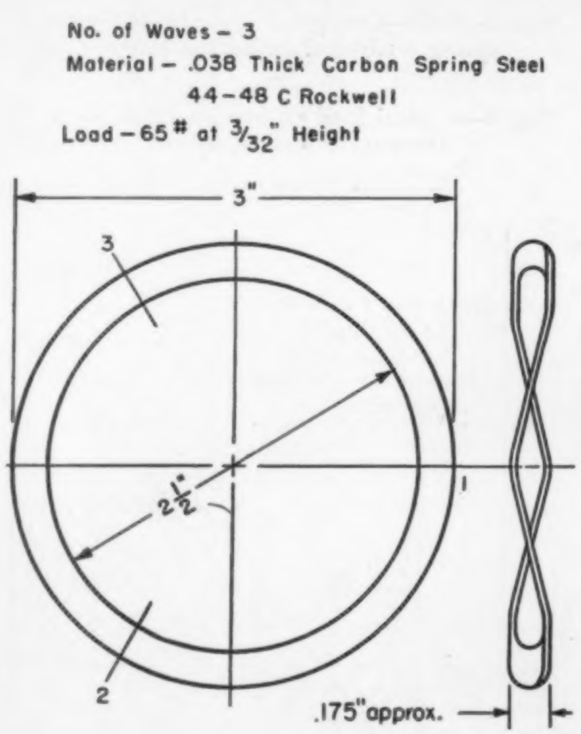


Fig. 3—Above—Drawing showing usual design specifications for a three-wave spring

Fig. 4—Below—Sketch of a four-wave spring giving formula symbols

where P = load, pounds; S = stress, psi; E = modulus of elasticity, psi; f = deflection, inches; b = radial width of material, inches; t = axial thickness of material, inches; N = number of waves; and D = mean diameter, inches, *Fig.* 4.

The following example will illustrate proper use of these calculations: Required, a wave washer to fit into a 3⅛-inch bore and over a 2⅜-inch shaft and to provide a 150-pound load with approximately 3/32-inch deflection.

Since the deflection is comparatively large for a spring of this type, we would use the most flexible of the wave washers, the 3-wave washer. We could also assume a 3-inch outside diameter and a 2.5-inch inside diameter to fit the given conditions. This would make the mean diameter 2.75 inches. Substituting these values in the load-deflection formula and solving for t,

$$t^3 = \frac{1.94PD^3}{EfbN^4}$$

$$t^3 = \frac{1.94 \times 150 \times 2.75^3}{30{,}000{,}000 \times 0.094 \times 0.250 \times 3^4} = 0.000106$$

$$t = 0.0474\text{-inch}$$

Then, the standard material to use is 0.049-inch thickness.

$$0.094 \times \frac{0.0474^3}{0.049^3} = 0.085\text{-inch deflection}$$

Thus, the correct deflection to get 150 pounds is 0.085-inch with 0.049-inch material. The total deflection should be something more than the 0.085-inch required to give the load, because a load at the solid height cannot be held with any uniformity. In order to determine how much deflection we can safely expect from this design we use the stress-deflection formula and solve for the deflection at a predetermined stress. Since we want to use 200,000 psi as a maximum, we can check for maximum deflection by making that substitution in the formula

$$f = \frac{\pi^2 D^2 S}{12EtN^2}$$

$$f = \frac{\pi^2 \times 2.75^2 \times 200{,}000}{12 \times 30{,}000{,}000 \times 0.049 \times 3^2} = 0.094\text{-inch}$$

This does not allow much deflection between the load and the solid height, which in a spring of these proportions should be at least 1/64-inch. However, the increase in stress in this case caused by adding this allowance is quite small and could be allowed, for:

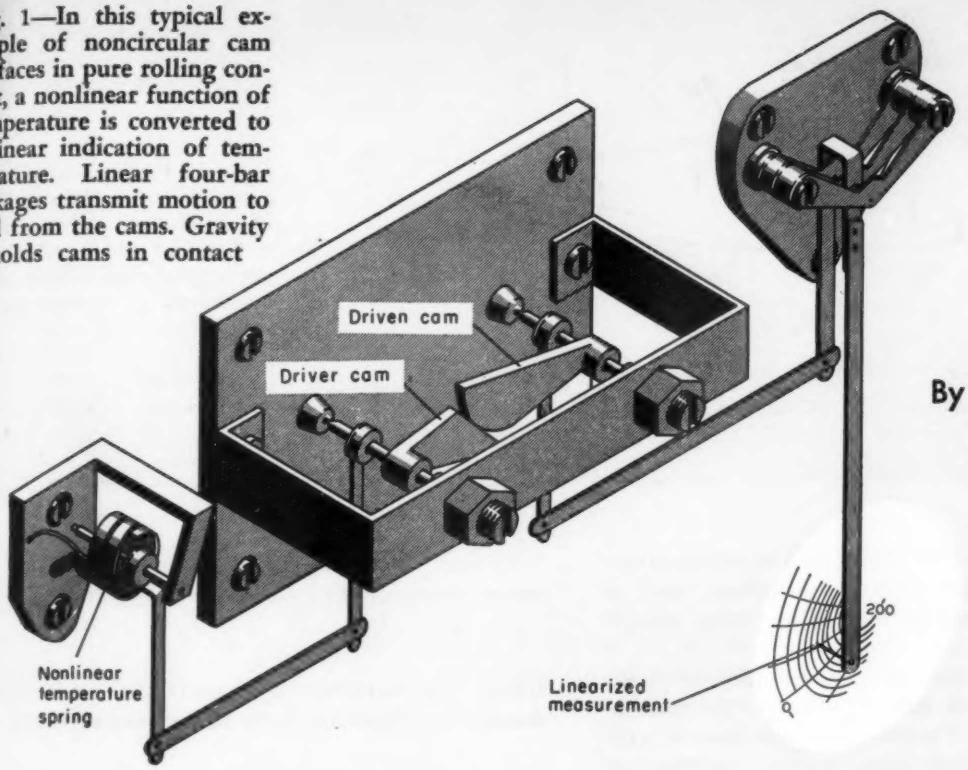
0.085 deflection required for load
0.016 additional deflection required
0.101-inch total deflection

and

$$0.101/0.094 \times 200{,}000 = 215{,}000 \text{ psi}$$

Wave washers are not meant to replace Belleville washers, nor are they a cure-all for problems of this type; however, they have their place in good design and can be a great help with proper application.

Fig. 1—In this typical example of noncircular cam surfaces in pure rolling contact, a nonlinear function of temperature is converted to a linear indication of temperature. Linear four-bar linkages transmit motion to and from the cams. Gravity holds cams in contact



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Designing Noncircular Surfaces

. . . for pure rolling contact

DESIGNING two rotating members to have one drive the other at an intentionally varying ratio is a problem that is frequently, but far from exclusively, encountered in measuring, controlling and computing equipment. The conversion in motion required may be from nonlinear to linear character, vice versa, or from one nonlinear motion to another found to be more practicable.

Examples requiring such conversions readily present themselves. In the measurement field it is often convenient to determine the value of a given variable indirectly by measuring a secondary variable which is mathematically related to the one in question. More often than not this association is of a nonlinear character, and if a uniform measurement scale is required, it becomes necessary to linearize the motion caused by this secondary variable. Common examples are the measurement of flow by means of the differential pressure existing across an orifice or other such applications of Bernoulli's theorem and measurement of temperature by determining the saturated vapor pressure exerted by a specific liquid in a closed system subjected to the temperature in question. Typical of this last example is the mechanism shown in *Fig.* 1.

Most proportional mechanisms in controlling instruments are linear throughout the entire range of measurement. With a nonlinear measurement scale, however, the behavior of the mechanism becomes different for each portion of the scale. Therefore, to obtain full advantage of the controlling mechanism, it may be desirable to convert the measurement to one in which the scale increments are equidistant.

In the design of computing equipment it is often found necessary to linearize the input to simplify the design of the computer mechanism. Characteristic of these problems are those encountered with the various types of continuous integrators.

Problems of this general nature can usually be solved with any of several elementary linkages or gearing mechanisms. However, when the function at hand is of a complicated nature, these mechanisms tend to become excessively cumbersome, from both structural and analytical viewpoints, and are often characterized by many moving parts with inherent

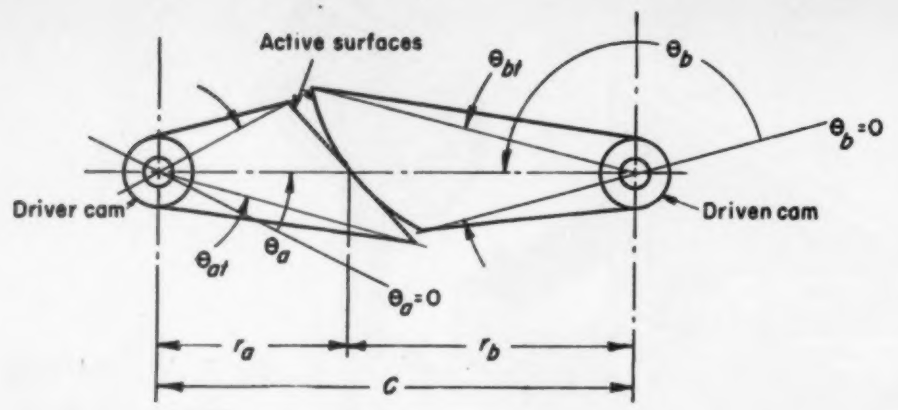


Fig. 2—Polar co-ordinates θ and r define the contours of driver and driven cams, and can be calculated to suit any appropriate relationship of driver to driven travel

and obvious inaccuracies. Advantages of accomplishing the required results with a mechanism that is both mathematically sound and structurally simple are apparent.

Particularly where the more elementary mechanisms are not applicable or are of an objectionable nature, the principles of noncircular surfaces in pure rolling contact are often found useful. Application of such surfaces often results in mechanisms embodying many desirable characteristics—minimum lost motion, few moving parts, and low friction. Problems of accuracy are dependent primarily upon the precision with which the theoretical can be reproduced in the model.

Graphical Methods Inadequate

Although the operation of such surfaces has long been understood and discussed in several of the standard mechanism texts, most of the treatments given are rather limited in scope and the solutions graphical in nature. Such discussions are usually restricted to parabolic, hyperbolic, elliptic, and logarithmic surfaces and are thereby limited in application.[1] Moreover, the graphical methods of construction are often tedious and insufficiently accurate for modern instruments.

This article presents a mathematical analysis together with an outlined method for producing required surfaces.[2] The mechanism under consideration is to consist of two plate cams, a driver and its driven, so machined that rotation of the driver will result, by means of surface contact, in the required mode of rotation of the driven cam. *Fig.* 2 illustrates the basic mechanism; symbols are defined in the Nomenclature.

By functional notation, the relationship between the angular positions of the driver and driven cams is

$$\theta_a = f(\theta_b) \tag{1}$$

By definition, for each cam

$$K = \frac{\theta_t}{P_{max} - P_{min}} \tag{2}$$

[1] References are tabulated at end of article.

Therefore, it is possible to express cam position in measurement units by

$$\theta = KP \tag{3}$$

Using this notation to express the relationship indicated in Equation 1 in measurement units

$$P_a = f(P_b) \tag{4}$$

For convenience, the cams in this particular development are to be so designed that the point of contact will always lie on the line of centers. Therefore,

$$r_a + r_b = C \tag{5}$$

Total lengths of the surfaces of contact on both cams will be the same, since they are in pure rolling contact. The total active surface length for the driver cam can be expressed by the equation

$$S = \int_0^{\theta_{at}} r_a d\theta_a \tag{6}$$

Correspondingly for the driven cam

$$S = \int_0^{\theta_{bt}} r_b d\theta_b \tag{7}$$

But since S is the same for both cams, and the limits are compatible

$$ds = r_a d\theta_a = r_b d\theta_b \tag{8}$$

Substituting from Equations 3, 4 and 5

$$r_a K_a df(P_b) = (C - r_a)\ K_b dP_b \tag{9}$$

Solving Equation 9 explicitly for r_a

$$r_a = \frac{CK_b dP_b}{K_b dP_b + K_a df(P_b)} \tag{10}$$

Dividing through by $K_b dP_b$

$$r_a = \frac{C}{1 + \dfrac{K_a df(P_b)}{K_b dP_b}} = \frac{C}{1 + \dfrac{K_a}{K_b} f'(P_b)} \tag{11}$$

where f' represents the first derivative of $f(P_b)$.

Rewriting Equation 5 explicitly for r_b

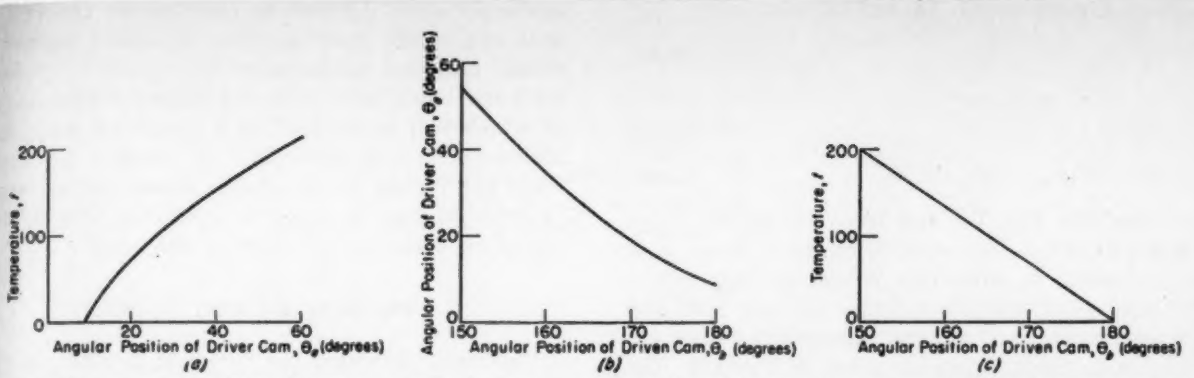


Fig. 3—These graphs show how cams transform a nonlinear function to a linear one. A variable, temperature, causes nonlinear displacement of the driver cam at *a*. What happens in the cam mechanism, angular position of driver versus driven, is shown at *b*. Finally, temperature versus angular position of the driven cam is shown at *c*

$$r_b = C - r_a \quad (12)$$

Solving for θ_a by substitution of Equation 3 in Equation 4

$$\theta_a = K_a P_a = K_a f(P_b) \quad (13)$$

And, rewriting Equation 3 in terms of the driven cam,

$$\theta_b = (180 - K_b P_b) \quad (14)$$

Equations 11, 12, 13 and 14 yield all the information necessary to construct any cams which are mathematically suitable to this application; that is, for cases in which $f'(P_b)$ is not equal to zero.

Example Shows Application

For illustration, the following example is presented to show how these equations can be used to solve a practical problem. The example, though hypothetical, is characteristic of the type in which this mechanism can be used with advantage.

Suppose that it has been found desirable to measure the temperature of a fluid indirectly by (1) measuring the saturated vapor pressure exerted by another fluid in a closed system subject to the fluid in question and (2) then converting this pressure measurement to a linear temperature scale. The saturated vapor pressure corresponding to the temperature has been found to be of the form $P = le^{mt/(t+n)}$. It is to be further assumed that this pressure can be converted to a uniform angular rotation of a shaft by means of a Bourdon tube or similar mechanism. Let the element be so designed that a 50 psi change in pressure (10 to 60 psi) will produce a 45.5-degree movement of the driver cam and that the full scale range of temperature (0 to 200) shall be linear over a 30-degree span. Assumed arbitrary constants for purposes of this problem are $l = 10$, $m = 6$ and $n = 470$. The center distance between the cams is chosen as $C = 3$ inches, since this has been found to be the largest size practical, consistent with the space limitations of the hypothetical problem. In the equation for vapor pressure, P is P_a of Equation 4 and $le^{mt/(t+n)}$ is $f(P_b)$.

The function to be performed by cams such as those shown in *Fig.* 2 is described graphically by the three curves of *Fig.* 3.

From Equation 2, $K_a = 45.5/(60 - 10) = 0.91$ and $K_b = 30/(200 - 0) = 0.15$.

From Equation 11,

$$r_a = \frac{3}{1 + \frac{0.91}{0.15}\frac{d}{dt}10e^{\frac{6t}{t+470}}}$$

By completing the differentiation,

$$r_a = \frac{3}{1 + 60.7\left[\frac{2820}{(t+470)^2}\right]e^{\frac{6t}{t-470}}} \quad (11a)$$

Table 1—Computed Cam Co-ordinates

t	r_a	θ_a	r_b	θ_b
0	1.69	9.1	1.31	180.00
25	1.54	12.3	1.46	176.25
50	1.41	16.2	1.59	172.50
75	1.29	20.8	1.71	168.75
100	1.20	26.1	1.80	165.00
125	1.11	32.1	1.89	161.25
150	1.03	38.8	1.97	157.50
175	0.97	46.4	2.03	153.75
200	0.92	54.1	2.08	150.00

Nomenclature

- a = Subscript designation for driver cam
- b = Subscript designation for driven cam
- C = Center distance of cams, in.
- K = Angle per measurement increment, degrees
- P = Position of cam corresponding to θ in measurement units (lb, psi, degrees, etc.)
- r = Instantaneous radius of cam corresponding to θ, in.
- S = Total length of surface of contact, in.
- θ = Instantaneous angle of cam with respect to $\theta = 0$, degrees
- θ_t = Total angular travel of cam (computed surface), degrees

Also, from Equations 12, 13 and 14

$$r_b = C - r_a = 3 - r_a \qquad (12a)$$

$$\theta_a = K_a f(P_b) = 9.1 e^{\frac{6t}{t+470}} \qquad (13a)$$

$$\theta_b = 180 - K_b P_b = 180 - 0.15t \qquad (14a)$$

Equations 11*a*, 12*a*, 13*a* and 14*a* yield all the information required for construction of these cams. It is simply necessary to substitute values of temperature *t*, into these equations to determine the cam radii and the corresponding cam angles. Numerical values obtained by such substitution are given in TABLE 1. The cams plotted from these values are the ones shown in *Fig.* 2. The computed co-ordinates given in TABLE 1 should, of course, be carried out to more significant figures in the actual application to obtain the required accuracy.

Cartesian Co-ordinates Aid Manufacture

Although Equations 11, 12, 13 and 14 are the only ones needed to compute the cam surfaces for any suitable set of conditions, certain refinements to these equations may be obtained by further mathematical manipulation. For example, if the cams are to be made with a jig borer to obtain the maximum accuracy, it may be desirable to convert the equations from polar to Cartesian co-ordinates. Even though the cam surfaces can be computed to any accuracy desired, it is obviously quite difficult to machine an accurate cam with the profile itself specified in either manner. Instead, practical advantages are realized by dealing with the center of a cutter of known radius, the edge of which will be tangent to a computed point on the active surface of the cam. Of course, for best results the radius of the cutter should be as large as practicable, but it must be less than the minimum radius of concave curvature of the cam.

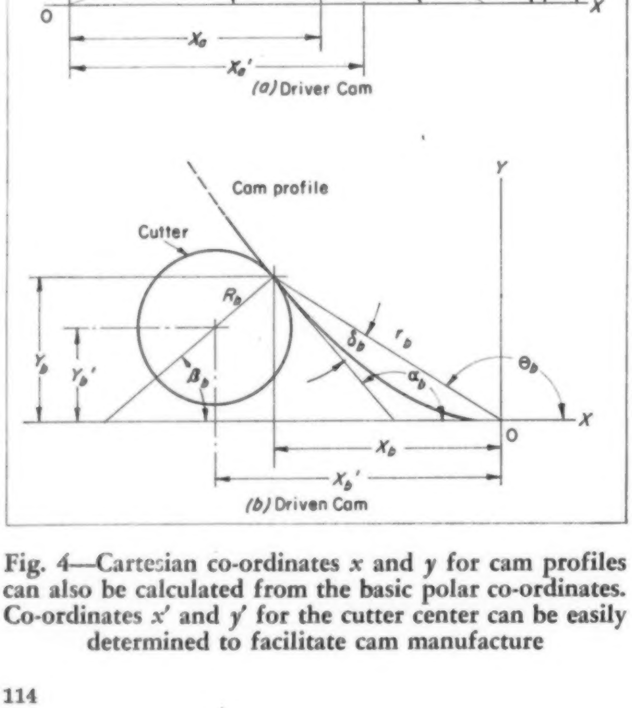


Fig. 4—Cartesian co-ordinates *x* and *y* for cam profiles can also be calculated from the basic polar co-ordinates. Co-ordinates *x'* and *y'* for the cutter center can be easily determined to facilitate cam manufacture

Determining Cutter Location

Geometry involving the cutter is detailed in *Fig.* 4. The center of the cutter must always lie on the normal to the curve at the point being determined, and out a distance equal to the radius of the cutter. The technique to be used is to develop a general equation for the slope of the cam in Cartesian co-ordinates. With the slope computable, the normal can be determined and the center of the cutter found.

Therefore, for the driver cam (*Fig.* 4*a*)

$$\tan \delta_a = \frac{r_a d\theta_a}{dr_a} \frac{\pi}{180} \qquad (15)$$

and from Equations 11 and 13

$$\frac{d\theta_a}{dr_a} = \frac{K_a f'(P_b) K_b \left[1 + \frac{K_a}{K_b} f'(P_b)\right]^2}{-CK_a f''(P_b)} \qquad (16)$$

Now substituting into Equation 15 from Equations 11 and 16

$$\tan \delta_a = \frac{r_a d\theta_a}{dr_a} \frac{\pi}{180} = -\frac{\pi K_b C f'(P_b)}{180 r_a f''(P_b)} \qquad (17)$$

From *Fig.* 4*a*

$$\alpha_a = \theta_a + \delta_a \qquad (18)$$

$$\tan \alpha_a = \frac{\tan \theta_a + \tan \delta_a}{1 - \tan \theta_a \tan \delta_a} \qquad (19)$$

Substituting known values into Equation 19

$$\tan \alpha_a = \frac{\dfrac{y_a}{x_a} - \dfrac{\pi K_b C f'(P_b)}{180 r_a f''(P_b)}}{1 + \dfrac{\pi K_b C y_a f'(P_b)}{180 r_a x_a f''(P_b)}} \qquad (20)$$

where

$$y_a = r_a \sin \theta_a \qquad (21)$$

$$x_a = r_a \cos \theta_a \qquad (22)$$

Therefore, since $\beta_a = \alpha_a - 90$, the angle of the normal can be expressed as

$$\beta_a = \tan^{-1} - \left[\frac{1 + \dfrac{\pi K_b C y_a f'(P_b)}{180 r_a x_a f''(P_b)}}{\dfrac{y_a}{x_a} - \dfrac{\pi K_b C f'(P_b)}{180 r_a f''(P_b)}}\right] \qquad (23)$$

The location of the center of the cutter is then

$$x_a' = x_a + R_a \cos \beta_a \qquad (24)$$

(*Continued on Page* 190)

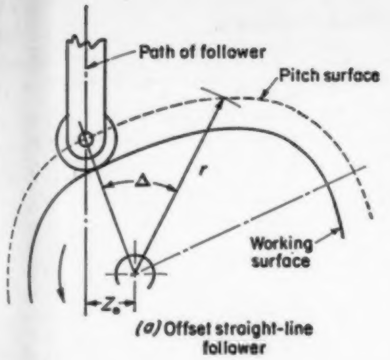
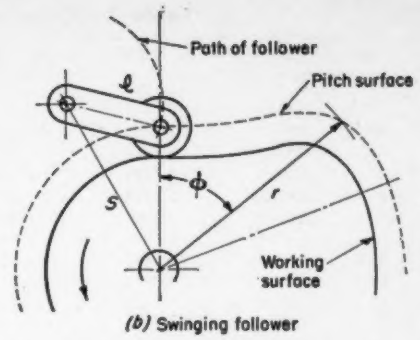


Fig. 1—Polar co-ordinates for points on cam pitch surface profiles with follower systems such as these can be easily calculated in tabular form



Calculating Cam Profiles

. . . a simplified method for determining co-ordinates of cam pitch surfaces

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IN THE many articles* that have appeared in MACHINE DESIGN and other publications, mathematics and application of various cam profile curves have been ably discussed. The objective of this article is to carry on from the point where a particular profile has been chosen and to describe a procedure for calculating co-ordinates of the pitch surface profile. The processes of making a master cam or leader from the profile data and the ability of modern production methods to cut a profile of required accuracy will also be discussed.

The master cam or leader is usually made in the form of a plate which has been accurately cut to the dimensions of the profile. This plate is then used as a guide in a suitable duplicating machine to position a cutter accurately in relation to a cam blank. If the master cam is made the same size as the finished cam the follower motion on the master can be applied directly to the motion of the cutter. In that event the diameter of the follower roller must be the same as the diameter of the cutter used in making the master cam. If extreme accuracy is needed, a master cam several times the size of the production cam can be used. This requires a suitable ratio between the follower motion and the cutter motion in the duplicating machine.

Consider first the making of a master cam for a typical face cam or a plate cam with a roller follower. Two types of follower motion will be considered: the straight-line motion follower, *Fig. 1a,* and the swinging follower whose path is a circular arc, *Fig. 1b.*

Customarily, in the manufacture of an accurate master cam, the profile is cut on a jig borer or an accurate milling machine with each point in the profile cut to a specified dimension in either polar or Cartesian co-ordinates. If a roller follower is to be used on the production cam, the master cam would be cut with the cutter axis parallel to that of the cam and a cutter diameter equal to the master cam follower roller. If a flat follower is to be used, an end mill type of cutter would be used with the plane of the cut parallel to the master cam axis.

For a master cam of the type being considered—plate or face cam—polar co-ordinates seem the logical choice and are used in this development, *Fig. 1.* The master cam blank is indexed to the specified angle

* See general references at end of article.

Table 1—Calculations for Cam Profile with Straight Offset Follower

θ	x_f	x_f+BC	$(x_f+BC)^2$	$(x_f+BC)^2+Z_0^2$	r	Z_0/r	δ	$\delta - \Omega$	Δ
0	0	2.8284	......	......	3.00000	0.3333	70.55	0	0
5	0.0178	2.8462	8.10085	9.10085	3.0170	0.3315	70.65	0.10	5.10
10	0.0712	2.8996	8.40768	9.40768	3.0670	0.3261	70.98	0.43	10.43
15	0.1600	2.9884	8.93053	9.93053	3.1513	0.3173	71.50	0.95	15.95
20	0.2848	3.1132	9.69201	10.69201	3.2698	0.3058	72.18	1.63	21.63
25	0.4440	3.2724	10.70860	11.70860	3.4218	0.2922	72.99	2.44	27.44
30	0.6408	3.4692	12.03534	13.03534	3.6104	0.2770	73.92	3.37	33.37
35	0.8710	3.6994	13.68556	14.68556	3.8322	0.2610	74.87	4.32	39.32
37.5	1.0000	3.8284	14.65665	15.65665	3.9568	0.2527	75.35	4.81	42.31
40	1.1290	3.9574	15.66102	16.66102	4.0932	0.2443	75.85	5.30	45.30
45	1.3592	4.1876	17.53599	18.53599	4.3053	0.2323	76.55	6.00	51.00
50	1.5560	4.3844	19.22296	20.22296	4.4970	0.2224	77.15	6.60	56.60
55	1.7152	4.5436	20.64430	21.64430	4.6523	0.2150	77.55	7.00	62.00
60	1.8400	4.6684	21.79395	22.79395	4.7743	0.2095	77.92	7.37	67.37
65	1.9288	4.7572	22.63095	23.63095	4.8612	0.2057	78.15	7.60	72.60
70	1.9822	4.8106	23.14187	24.14187	4.9134	0.2035	78.24	7.69	77.69
75	2.0000	4.8284	23.31345	24.31345	4.9309	0.2028	78.30	7.75	82.75

Equations and constants: x_f = selected function of θ; $r = \sqrt{(x_f + BC)^2 + Z_0^2}$; $\cos \Omega = Z_0/R = 1/3$; $\Omega = 70.55$; $\cos \delta = Z_0/r$; $\gamma = \delta - \Omega$; $\Delta = \theta + \gamma$

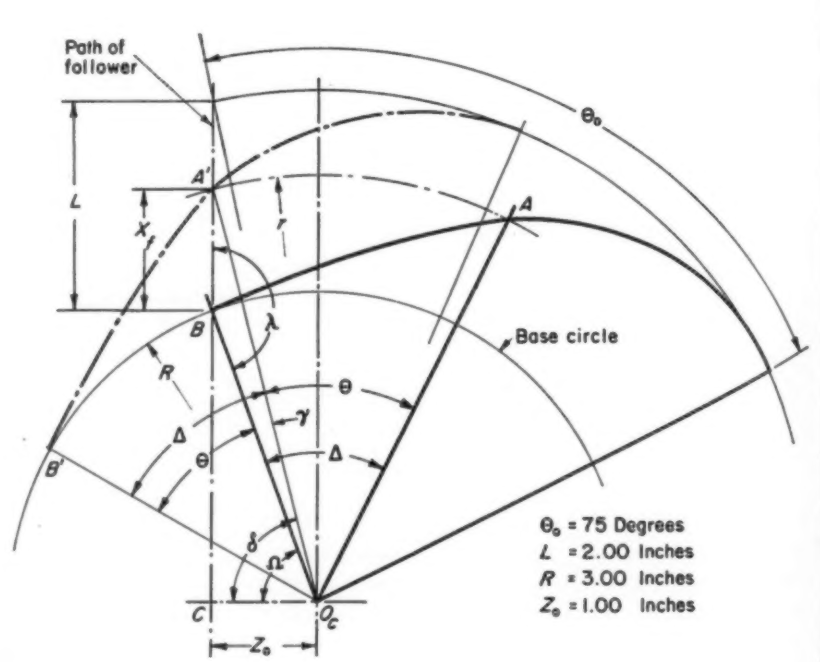


Fig. 2—This layout gives the essential properties employed in the tabular computation of Δ and r in Table 1 for an offset straight-line follower system. The pitch profile is shown advanced counterclockwise through angle θ with points A and B becoming A' and B'

and the center of the cutter fed into the blank radially to the radius specified for the pitch surface. The master cam as it comes from the jig borer has a profile which consists of a series of small scallops if a roller follower is to be used, and a series of small flat spots if a flat follower is contemplated. The true working profile is the curve which is tangent to the arcs of the scallops or the flats and the master cam profile must be hand finished to this curve. The final accuracy of the profile depends therefore on the craftsmanship of this final operation. The smaller the scallops or flats, the easier will be the hand finishing and consequently the higher the expected accuracy.

In *Fig.* 2 are shown diagrammatically the essential elements and dimensions of the cam profile laid out for an offset straight-line follower. *Fig.* 3 shows the same for a swinging follower. Symbols on these illustrations are defined in the Nomenclature.

Consider first the offset straight follower, *Fig.* 2. Since polar co-ordinates are to be used, the problem resolves itself into a determination of the values r and Δ for particular values of θ. From *Fig.* 2 it is obvious that

$$\Delta = \theta + \gamma \qquad (1)$$

Therefore if γ is evaluated, Δ will be easy to calculate. γ is a function of δ in the equation

$$\gamma = \delta - \Omega \qquad (2)$$

in which Ω is a constant for the particular geometry of the cam under consideration. The triangle BCO_c remains fixed and the value of Ω can be established by the equation

$$\cos \Omega = \frac{CO_c}{BO_c} = \frac{Z_0}{R} \quad \text{(3)}$$

The value of δ is a function of the profile radius r and can be found from the equation

$$\cos \delta = \frac{Z_0}{r} \quad \text{(4)}$$

It remains therefore to establish the value of the profile radius r for particular values of θ. This is most easily accomplished by considering the right triangle $A'CO_c$ and solving for r in the equation

$$r = \sqrt{(A'C)^2 + (CO_c)^2} \quad \text{(5)}$$

For the particular geometry of the cam under consideration Equation 5 can be written

$$r = \sqrt{(x_f + BC)^2 + Z_0^2} \quad \text{(6)}$$

in which BC and Z_0 are constant and x_f varies with cam angle according to the kind of profile chosen.

Tabulation Simplifies Calculations

In order to carry out the calculations in an orderly fashion it is expedient to tabulate the results. A sample solution for a cam of the dimensions shown in *Fig.* 1 has been tabulated in TABLE 1. The computations were made on an eight-digit calculating machine and the figures are accurate to the number of places shown.

As has already been mentioned, the accuracy of the final profile depends on the number of profile radii calculated or in effect upon the size of the increment between successive values of θ. It is customary to calculate values of r for every degree of cam motion θ and, for some high-speed cams, with even smaller increments.

In the solution for the profile radius r of the straight follower it is also possible to use the cosine law. The equation becomes

$$r = \sqrt{R^2 + x_f^2 - 2Rx_f\cos\lambda} \quad \text{(7)}$$

in which the angle λ is found as follows

$$\lambda = \Omega + 90 \quad \text{(8)}$$

This method seems more laborious since it involves the use of trigonometry tables.

Considering the swinging follower of *Fig.* 3 in terms of polar co-ordinates, the values of r and ϕ must again be related to known elements in the geometry of the cam in order to evaluate them for particular values of θ.

$$\phi = \theta + \gamma \quad \text{(9)}$$

In this case γ is a function of the angle σ according to the relation

$$\gamma = \sigma - \beta \quad \text{(10)}$$

Nomenclature

(Units: angles, degrees; lengths, inches)

l = Swinging follower lever radius
L = Cam lift
O_f = Center of rotation of swinging follower lever
O_c = Center of rotation of cam
r = Distance from center of rotation of cam to pitch surface
R = Radius of base circle of cam
S = Distance between center of rotation of follower lever and center of rotation of cam
x_f = Displacement of cam follower
Z_0 = Offset of straight follower from cam center
α = Angle between swinging follower centerline at $\theta = 0$ and line of centers of follower lever and cam
β = Angle between cam radius at $\theta = 0$ and line of centers of follower lever and cam
γ = Angular increment (plus or minus) from profile radius r at angle $\theta = 0$ to profile radius r at any angle θ
Δ = Angular displacement of profile radius r for straight follower at cam displacement θ
δ = Angle between profile radius r at A' position and horizontal center line through O_c
ϵ = Angle between centerline of follower lever and line of centers of follower lever and cam (The angle faced by line $A'O_c$)
θ = Angular displacement of cam
θ_0 = Angular displacement of cam required to produce full travel (lift) of follower
λ = Angle between profile radius r at $\theta = O$ and centerline of follower
σ = Angle between profile radius r and line of centers of follower lever and cam
ϕ = Angular displacement of profile radius r for swinging follower at cam displacement θ
ψ = Angular displacement of swinging follower lever at a particular value of θ (Corresponds to x_f in straight motion)
ψ_0 = Angular displacement of swinging follower lever at θ_0 (Corresponds to L in straight motion)
Ω = Angle between profile radius r at $\theta = O$ and offset dimension Z_0

in which β is constant and is defined as

$$\cos\beta = \frac{S^2 + R^2 - l^2}{2SR} \quad \text{(11)}$$

The profile radius r is a function of the angle ϵ in the triangle $A'O_fO_c$. In turn, ϵ is a function of the angle ψ in the relation

$$\epsilon = \psi + \alpha \quad \text{(12)}$$

in which α is a constant defined as

$$\cos \alpha = \frac{l^2 + S^2 - R^2}{2lS} \quad \text{(13)}$$

and ψ is the angular displacement of the swinging follower lever for a particular cam angle θ according to the particular cam profile chosen.

Calculations for a cam with a swinging follower and the dimensions shown in *Fig.* 3 have been tabulated in TABLE 2.

In *Figs.* 1, 2 and 3 cams with counterclockwise ro-

tation are shown. It is not the intention to show here all the possible variations of followers and cams but rather to suggest a method that can be applied to any combination.

For these sample calculations, a parabolic function was arbitrarily chosen as the source for values of x_f, *Fig.* 2, and ψ_0, *Fig.* 3. Values of θ from zero to 75 degrees were used in 5-degree increments for getting the values of r, ϕ, and Δ. The resulting cam pitch profiles are plotted on their respective *Figs.* 2 and 3.

Data furnished to the shop for making the master cam consist therefore of a drawing showing the details of the hub required and the profile outline, and a complete tabulation of θ, r, and ϕ or Δ for each profile on the cam. The thickness or face width of a master cam depends to some extent on the type of duplicating machine it will be used on and the number of cams that must be produced from it. Master cams vary from as thin as 1/16-inch in soft steel, where only a few cams are to be cut, to widths of ½ to ¾-inch in heat-treated steel for mass production.

Accuracy requirements in a master cam can best be emphasized by the figures in TABLE 3. Here are tabulated the displacements of a straight radial follower for three commonly used profiles: parabolic or "gravity," cubic, and simple harmonic. Base circle radius and other data are the same as for *Figs.* 2 and 3. Note the relatively small differences in the dis-

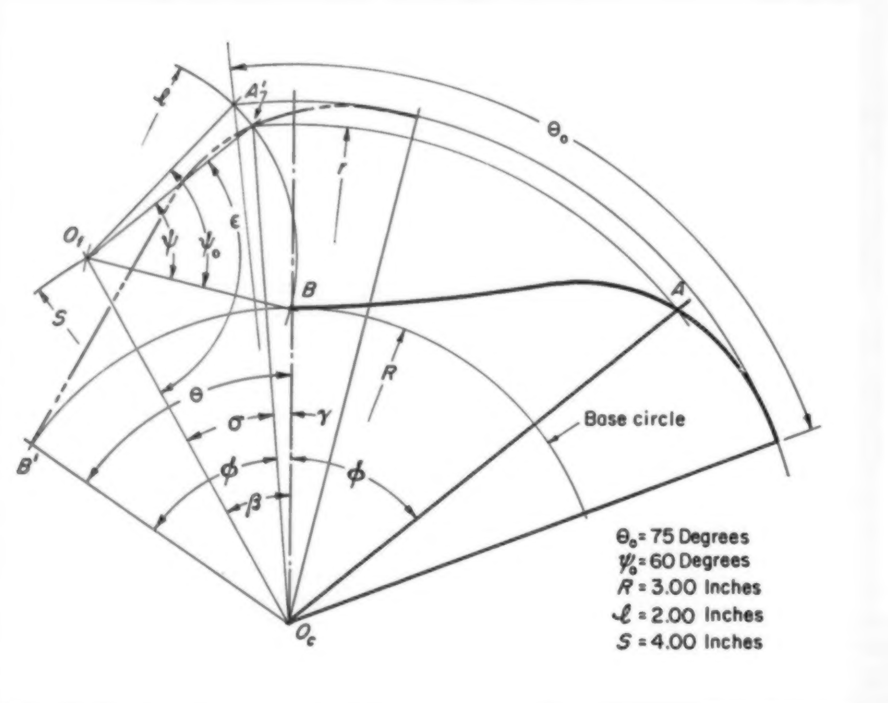


Fig. 3—Similar to *Fig.* 2, this layout shows the details used in the calculations of co-ordinates ϕ and r in Table 2 for a swinging follower system

Table 2—Calculations for Cam Profile with Swinging Follower

θ	ψ	$\psi+\alpha=\varepsilon$	$\cos\varepsilon$	$-2lS\cos\varepsilon$	r^2	r	$r^2+S^2-l^2$	$2Sr$	$\cos\sigma$	σ	γ	ϕ
0	0	......	.....	....	......	3.0000	......	......	......	....	...	....
5	0.5328	47.1021	0.6806	−10.8896	9.1104	3.0183	21.1104	24.1464	0.87426	29.04	0.08	5.08
10	2.1336	48.7029	0.6612	−10.5792	9.4208	3.0693	21.4208	24.5544	0.87238	29.28	0.32	10.32
15	4.8000	51.3693	0.6243	−9.9888	10.0112	3.1640	22.0112	25.3120	0.86969	29.59	0.63	15.63
20	8.5332	55.1025	0.5721	−9.1536	10.8464	3.2934	22.8464	26.3472	0.86713	29.88	0.92	20.92
25	13.3333	59.9026	0.5015	−8.0240	11.9760	3.4606	23.9760	27.6848	0.86603	30.00	1.04	26.04
30	19.2000	65.7693	0.4104	−6.5664	13.4336	3.6652	25.4336	29.3216	0.86740	29.84	0.88	30.88
35	26.1333	72.7026	0.2973	−4.7568	15.2432	3.9042	27.2432	31.2336	0.87224	29.39	0.43	35.43
37.5	30.0000	76,5693	0.2322	−3.7512	16.2488	4.0310	28.2488	32.2480	0.87599	28.84	0.12	37.62
40	33.8667	80.4360	0.1657	−2.6510	17.3490	4.1652	29.3490	33.3216	0.88078	28.26	−0.70	39.30
45	40.8000	87.3693	0.0459	−0.7340	19.2660	4.3893	31.2660	35.1144	0.89040	27.07	−1.89	43.11
50	46.6667	93.2360	−0.0565	+0.9042	20.9042	4.5721	32.9042	36.5768	0.89959	25.90	−3.06	46.94
55	51.4667	98.0360	−0.1499	2.3978	22.3978	4.7326	34,3978	37.8608	0.90853	24.70	−4.26	50.74
60	55.2000	101.7600	−0.2038	3.2608	23.2608	4.8229	35.2608	38.5832	0.91389	23.95	−5.01	54.93
65	57.8664	104.4357	−0.2494	3.9904	23.9904	4.8980	35.9904	39.1840	0.91850	23.29	−5.67	59.33
70	59.4672	106.0365	−0.2763	4.4208	24.4208	4.9417	36.4208	39.5336	0.92126	22.89	−6.07	64.93
75	60.0000	106.5693	−0.2851	4.5616	24.5616	4.9560	36.5616	39.6480	0.92215	22.75	−6.21	68.79

Equations and constants: ψ = selected function of θ; $\cos\alpha = (l^2 + S^2 - R^2)/(2lS) = 0.6875$; $\alpha = 46.5693$; $r^2 = l^2 + S^2 - 2lS\cos\varepsilon = 20 - 16\cos\varepsilon$; $\cos\sigma = (r^2 + S^2 - l^2)/(2rS) = (r^2 + 12)/8r$; $\cos\beta = (S^2 + R^2 - l^2)/(2SR) = 0.8750$; $\beta = 28.96$; $\gamma = \sigma - \beta$; $\phi = \theta + \gamma$

Table 3—Comparative Displacements for Radial Follower Systems

Cam Rotation θ (degrees)	Follower Displacement x_f (inches) for $\theta_o = 75$ degrees, $L = 1.5$ inches		
	Parabolic	Cubic	Harmonic
1	0.00053	0.00001	0.00068
2	0.00213	0.00011	0.00263
3	0.00480	0.00038	0.00593
4	0.00853	0.00091	0.01050
5	0.01333	0.00178	0.01643
6	0.01920	0.00307	0.02355
7	0.02613	0.00488	0.03203
8	0.03413	0.00728	0.04170
9	0.04320	0.01037	0.05263
10	0.05333	0.01422	0.06488

placements of the parabolic and the harmonic for the first two or three degrees. Minor errors in machining could make the profile one or the other, or neither. This is perhaps not so serious because the accelerations of the gravity and the harmonic profiles at the starting point, $\theta = 0$, are of the same order of magnitude. If on the other hand a combination curve using the cubic at the start were contemplated, extreme accuracy would be required in the first few degrees to insure the zero starting acceleration and the proper slope of the acceleration curve. This would also be true of the cycloidal curve. It is perhaps because of this almost impossible accuracy requirement that cubic and cycloidal curves have not been used to as great an extent as theory would indicate they should be. As manufacturing equipment improves and the technique of cutting accurate master cams is developed to higher accuracy, these desirable profile curves will have more frequent acceptance.

REFERENCES

G. T. Chapman—"Parabolic Cams Smooth Velocity Curves," MACHINE DESIGN, Feb., 1936, Pages 39-40.

W. H. Shorter Sr.—"Cam Design Analysis," *Product Engineering*, May 1940, Pages 223-227, and June, 1940, Pages 264-265.

W. M. Dudley—"A New Approach to Cam Design," MACHINE DESIGN, July, 1947, Pages 143-148 and 184.

John A. Hrones—"Key Factors in Cam Design and Application," MACHINE DESIGN, April, 1949, Pages 127-132; May, 1949, Pages 107-111 and 178; June, 1949, Pages 124-126.

D. B. Mitchell—"Cam Follower Dynamics," MACHINE DESIGN, June, 1950, Pages 151-154.

A. S. Gutman—"Cam Dynamics," MACHINE DESIGN, March, 1951, Pages 149-151, 202, 204.

Hot Workability of Alloy Steels Increased

INCREASED hot workability of high-alloy, corrosion-resistant steels is a result of a recent development of The Carpenter Steel Co. in which cerium is added to the alloying mixture. At least seven years in the development stage, the invention, U. S. patent No. 2,553,330, applies to ferrous alloys containing nickel and one or more of the elements chromium, molybdenum, cobalt, copper, tungsten, silicon, manganese, columbium and vanadium.

Heretofore, a principal obstacle to the full utilization of advantages offered by these alloys has been the difficulty or practical impossibility of forging and rolling them into desirable shapes such as sheets, rods and tubes. This obstacle has existed for years and has restricted the uses of these alloys in certain applications involving high temperatures, stresses and pressures. For example, certain alloys containing nickel, chromium, molybdenum and copper which provided satisfactory corrosion resistance could be cast satisfactorily—but when hot working was attempted, the resulting bars were so badly torn or cracked that they were unusable.

This development involves the proper application of a rare-earth element, cerium—an element used extensively in flints to produce sparks in ordinary cigarette lighters. Cerium is used by itself or in combination with its sister element, lanthanum, both found in misch metal, to produce hot workable alloys for many applications such as heat resistant baffles and shields, heat resistant parts for jet and turbo-jet aircraft, heat and corrosion resistant valves for internal combustion engines.

Pure cerium or pure lanthanum may be used separately, but combinations of the two metals within certain proportion limits have been found most effective. The nickel content of any of the alloys of this invention determines the minimum and maximum allowable ranges of cerium and lanthanum which will produce hot workability.

The Carpenter corrosion and heat resistant grades included in the new cerium-bearing alloy range include stainless AISI Types 309, 310, 316, 317, 330, as well as Carpenter Stainless 20, and Carpenter austenitic valve steels. The invention applies not only to ferrous alloys containing more than 50 per cent iron, but also to nonferrous alloys containing little or no iron. The accompanying illustration shows the standard cone test made for hot forgeability. The two pancakes are of the same analysis, except that the one on the left contains cerium.

Laminated-Molded Plastics

. . . practical for large and complex shapes

Official Department of Defense Photo

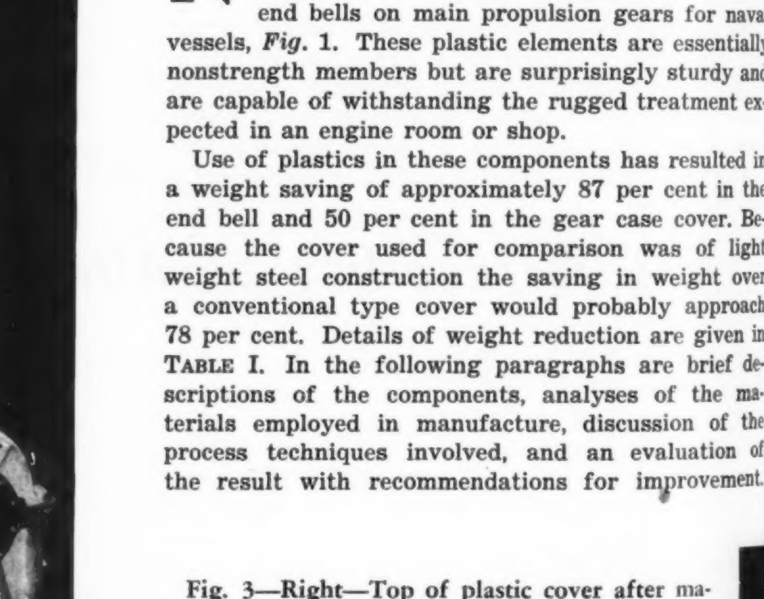
Fig. 1—Above—Plastic cover on an experimental main propulsion gear for advanced type naval vessels

Fig. 2—Right—Interior of plastic cover after removal from mold and prior to the machining operations

NEW MATERIALS and manufacturing techniques have made possible plastic covers and end bells on main propulsion gears for naval vessels, *Fig.* 1. These plastic elements are essentially nonstrength members but are surprisingly sturdy and are capable of withstanding the rugged treatment expected in an engine room or shop.

Use of plastics in these components has resulted in a weight saving of approximately 87 per cent in the end bell and 50 per cent in the gear case cover. Because the cover used for comparison was of light weight steel construction the saving in weight over a conventional type cover would probably approach 78 per cent. Details of weight reduction are given in TABLE I. In the following paragraphs are brief descriptions of the components, analyses of the materials employed in manufacture, discussion of the process techniques involved, and an evaluation of the result with recommendations for improvement.

Fig. 3—Right—Top of plastic cover after machining of joint surfaces



Photo, courtesy Winner Manufacturing Co.

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DESCRIPTION OF COMPONENTS: The gear cover, *Figs.* 2 and 3, was designed to be used on an experimental gear drive. This cover is not of uniform thickness throughout but for the most part is about ½-inch. Sections in the vicinity of inspection covers, thermocouple wells and flanges vary between ¼ and 1⅛ inches. The inspection ports have a ½-inch keyed-in aluminum insert around the periphery. This insert is drilled and tapped for securing the plastic inspection plates. The thermocouple wells are standard pipe coupling with flanges which are imbedded in the plastic at the time of fabrication.

End plates for the cover are fastened to the vertical plastic flanges by cap screws, *Fig.* 1. These screws are threaded into the plastic. Originally studs were used, *Fig.* 3. However, wear of the threads in the plastic prompted use of cap screws.

The end bell cover, *Fig.* 4, which is used over the coupling end of a destroyer escort main propulsion gear is fibrous glass impregnated polyester resin approximately ½-inch thick.

MATERIALS AND FABRICATION: The material used has low density and good physical strength, resists hot lubricating oil, and appears to be economical from the standpoint of manufacture. This resin, when reinforced with glass fibres in the form of resin bonded mats, woven cloths, or combination of both, results in high strength laminates capable of being fabricated into large and complex shapes. Typical physical properties obtained from tests on a ½-inch laminated section are listed in TABLE 2.

In the selection of a resin for this application, tensile strength tests were conducted on polyester resin glass laminates after various periods of immersion in oil to determine possible losses in physical strength. Results of these tests determined the materials selected.

The gear case cover was manufactured by a vacuum bag process using a plastic male mold and a rubber bag to obtain pressure by means of evacuating the bag surrounding the object. The entire assembly was then cured by oven heat. The surfaces of the gear cover were oilproofed by means of a technique developed at the US Naval Ordnance Laboratory. The method involves using a low viscosity one hundred per cent activated, heat curable liquid resin as a wet sanding medium and heat curing a very light coating of the liquid resin on the surface. This treatment filled the minute pores not discernible to the naked eye. Present day fabricating techniques utilizing these materials are capable of producing pieces having the required oil resistance without subsequent surface treatment.

The end bell cover was manufactured by the US Naval Ordnance Laboratory, White Oak, Md. The

Official Department of Defense Photo

Table 1—Gear Cover and End Bell Data

	Dimensions (in.)	Weight (lb)
Gear Cover		
Conventional steel construction	60 x 50 x 12	300–350*
Lightweight steel	60 x 50 x 12	163
Plastic	60 x 50 x 12	77
End Bell		
Steel	24 dia. x 10†	135
Plastic	24 dia. x 10†	18

* This is an estimated weight.
† The 24-inch diameter is the flange OD. Bell diameter is 20 inches. Flange thickness is ½-inch. Bell depth overall is 10 inches.

Table 2—Typical Plastic Laminate Properties

Tensile strength	27,000 psi
Flexural modulus	11.2 x 10⁵ psi
Flexural strength	31,000 psi
Compressive strength	44,300 psi
Impact strength (Izod, edgewise)	26 ft-lb/in. notch
Rockwell hardness	109M
Specific gravity	1.64
Fatigue strength (5 x 10⁶ cycles)	7,400 psi

piece was made by laying polyester resin impregnated fibrous glass cloth on a female mold. A rubber bag was placed over the entire piece and evacuated. The entire piece was then placed in a steam autoclave and cured at 30 psi gage. This piece was not treated for oil resistance. Its appearance was enhanced by coating the outside with phenolic resin varnish pigmented with metallic aluminum.

Machining: Generally, this material may be easily machined if proper precautions are observed. High speed tools are usually required where only a small amount of machine work is to be done. Carbide-tipped tools are utilized where a large amount of cutting is necessary. Standard metal working lathes, shapers, planers, millers, drills, etc., may be employed to machine this plastic.

Testing: The plastic gear case cover and end bell have been installed on full size reduction gear units under test at the US Naval Boiler and Turbine Laboratory, Philadelphia. Tests of the gear case cover varied from no load to full load and extended over a period of approximately 180 hours. Forty-five hours were under load carrying conditions and the remainder at light or no loads. During this period the cover was removed and reinstalled five times to permit inspection and modification of the gear elements. The oil supplied to the reduction gear was Navy Symbol 2190-T. The rate of supply was approximately 40

Fig. 4—Plastic end bell cover for a destroyer escort main propulsion reduction gear

Official Department of Defense Photo

gallons per minute at 125 F. A portion of this oil was thrown from the tooth mesh and splashed against the under side of the plastic cover. This plastic cover was inspected during various periods of the test and there was no indication of failure such as cracks or chipping at any point. Oil absorption appeared to be negligible. There was no deterioration of the cover at the area where the lubricating oil contacted the plastic.

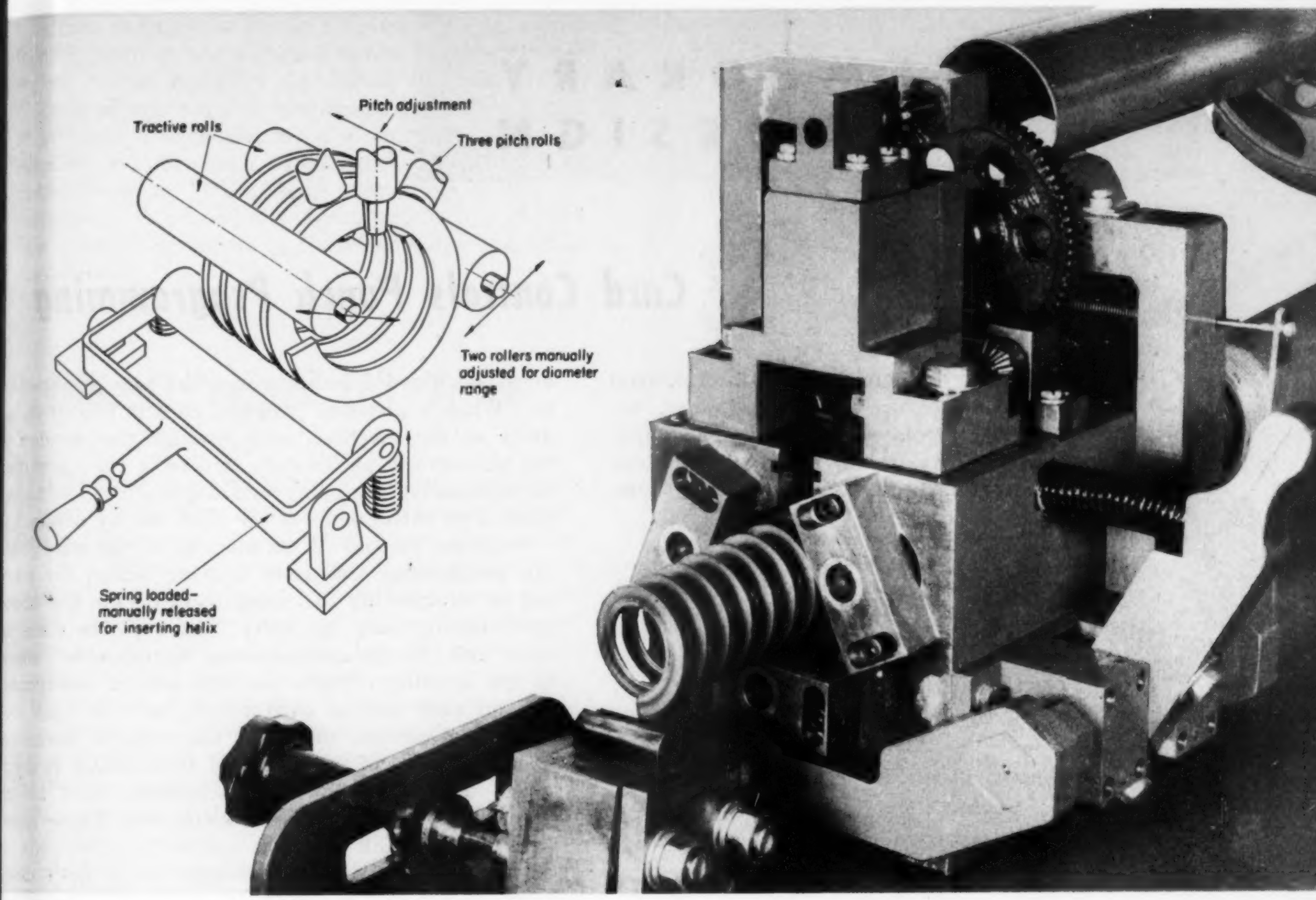
The end bell cover was tested on a destroyer escort reduction gear operated under varying loads. The actual running of the gear with this plastic element in place was approximately 60 hours. Of this running time 18 hours were under actual load carrying conditions. The oil supplied to the gear was Navy Symbol 2190-T. The end bell cover was installed and removed twice during test. Vibrations encountered were negligible. There was no sign of any distress, such as fatigue cracks or attack by oil on the surface of the plastic.

Suggestions for New Designs: The gear case cover and end bell are considered very satisfactory from the standpoint of savings in weight and for operation. However, for new designs a number of points should be considered:

1. Metal inserts should be placed in the plastic where studs or cap screws subjected to frequent removal are to be used. These inserts may be incorporated in the piece during fabrication. The inserts may be drilled to suit
2. Where through bolts are to be used, metal bushings would be desirable where removal is frequent. These bushings could be placed in the mold as inserts at the time of fabrication and drilled to suit
3. A polyester resin used in conjunction with a resin bonded glass mat is suitable for applications of this type. This resin has been satisfactory from the standpoint of oil absorption and also resistance to failure when subjected to excessive vibration
4. Manufacture of similar items can involve techniques other than the vacuum bag method. The use of matched tooling, vacuum-injection method with room temperature curing polyester resins would produce an equally satisfactory piece
5. Use of a high strength, synthetic cellular material as a core between rigid skins suggests the achievement of high strengh, light weight, plus sound and vibration attenuation features

Conclusions: These plastic components have demonstrated attractive features in applications relating to shipboard uses. One of the most outstanding advantages is the saving of weight. The large reduction in weight is also desirable from the standpoint of maintenance. The parts described are of such weight that they can easily be handled without the use of chainfalls while the steel counterparts could not be installed or removed without use of heavy equipment. The ease of manufacture and the use of noncritical materials is not to be ignored in designing machinery parts for the applications described here or for parallel industrial uses.

The writers wish to thank the Bureau of Ships, Department of Defense, for permission to publish this paper. Assistance of personnel at the US Naval Boiler and Turbine Laboratory, US Naval Ordnance Laboratory and the Bureau of Ships, in collecting and reviewing this information, is greatly appreciated.



Mill Corrects Helix Pitch

By R. E. Blaney
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Westinghouse Electric Corp.
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DESIGNED to adjust the pitch of springs or helical resistor elements to any desired number of turns per inch, the miniature mill shown in the photograph above employs three formed rolls to apply the necessary bending action on the wire section. Tightwound preformed helices are driven through the pitch adjustment rolls by three traction rolls which also serve as straightening rolls, drawing, left above.

To facilitate loading into the mill and to apply uniform traction force, the lower of the three traction rolls is spring loaded. This also provides a limited capacity for differing outside diameters. Driven by the same power shaft, all the rolls are synchronized automatically.

To permit adjustment of the amount of pitch change effected, the pitch rolls are adjustable and the upper moveable housing is provided with a calibrated scale. Angular position of the rolls conforms to the radial shape of the wire processed and, since the rolls are required to operate between the turns of the helix, roll diameter must be held within 5/16-inch.

Integral with the pitch mill is a shear so designed that its blades can be inserted between the turns of the helix which it cuts. The blades are sufficiently thin to enter between turns of the most closely wound helix, yet they have sufficient strength to withstand a force of 4000 pounds at their extremities.

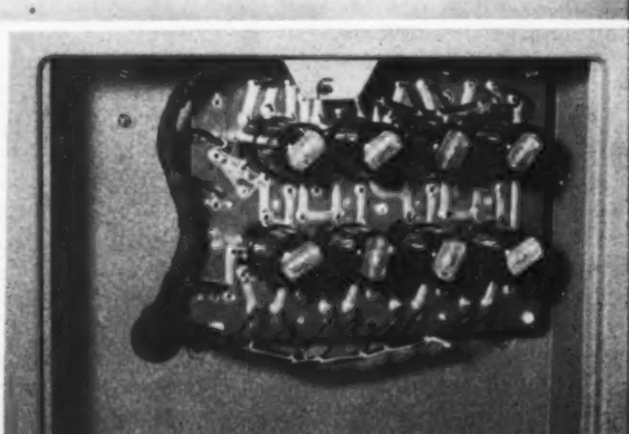
Although it is highly desirable to set the pitch of any such accurate helical wire form at the time of winding, varying physical properties, different ribbon-wire sizes used, and numerous pitch requirements create prohibitive tooling costs. Also, owing to the stiffness of most wires employed, assembly without accurate pitch is often impractical, especially where resistor elements are to be mounted into porcelain supports. With this machine for adjusting pitch to suit requirements, preliminary forming into helical shape on one set of standard winding rolls can be readily utilized with economical results.

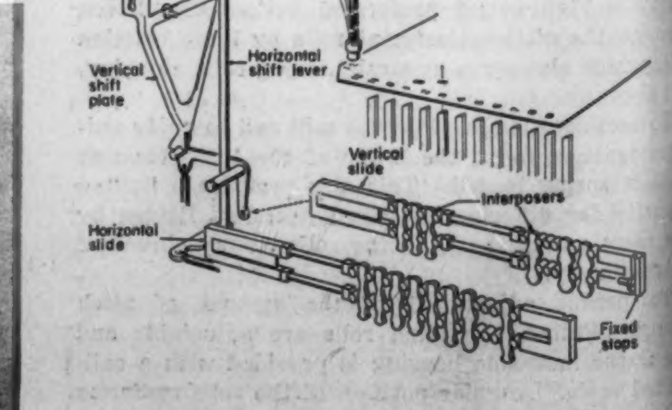
CONTEMPORARY DESIGN

Card Controls Punch Programming

CARD control of programming in the International Business Machines Corp. electric card punch, below, automatically controls card skipping and duplicating, thus eliminating skip bars and tabular inserts. Card programming also directs whether the combined keyboard punches numerically or aphabetically. When a prepared program card is placed on the drum of the machine, seen through the window of the photograph at the left, it directs the operation automatically. Skipping and duplicating are accomplished by entire fields in the card, not by columns.

A unique feature of the machine is the wire printing mechanism, schematic drawing below, for printing or interpreting the coded holes on the top of the card directly over the holes. As a punch rises, its lower end lifts its corresponding "interposers", shown in the drawing. There are two sets of interposers, one on each side of each punch, with one set controlling a vertical slide and the other a horizontal slide. Amount of movement of these slides is determined by the size of the interposers lifted by the particular punch that is in operation. These slides, through bellcrank linkages, operate a code plate by moving it vertically, horizontally, or in both directions. When the code plate is positioned, a lever (not shown) moves the code plate forward to contact some of the 35 wires projecting from a wire guide and funnelled to a small form, over the top of the card, which is the overall size of a letter. The code plate is designed with projections so that, when it is shifted to





a certain position as outlined previously and moved forward, appropriate wires are moved to press against a cloth ribbon and print the correct character.

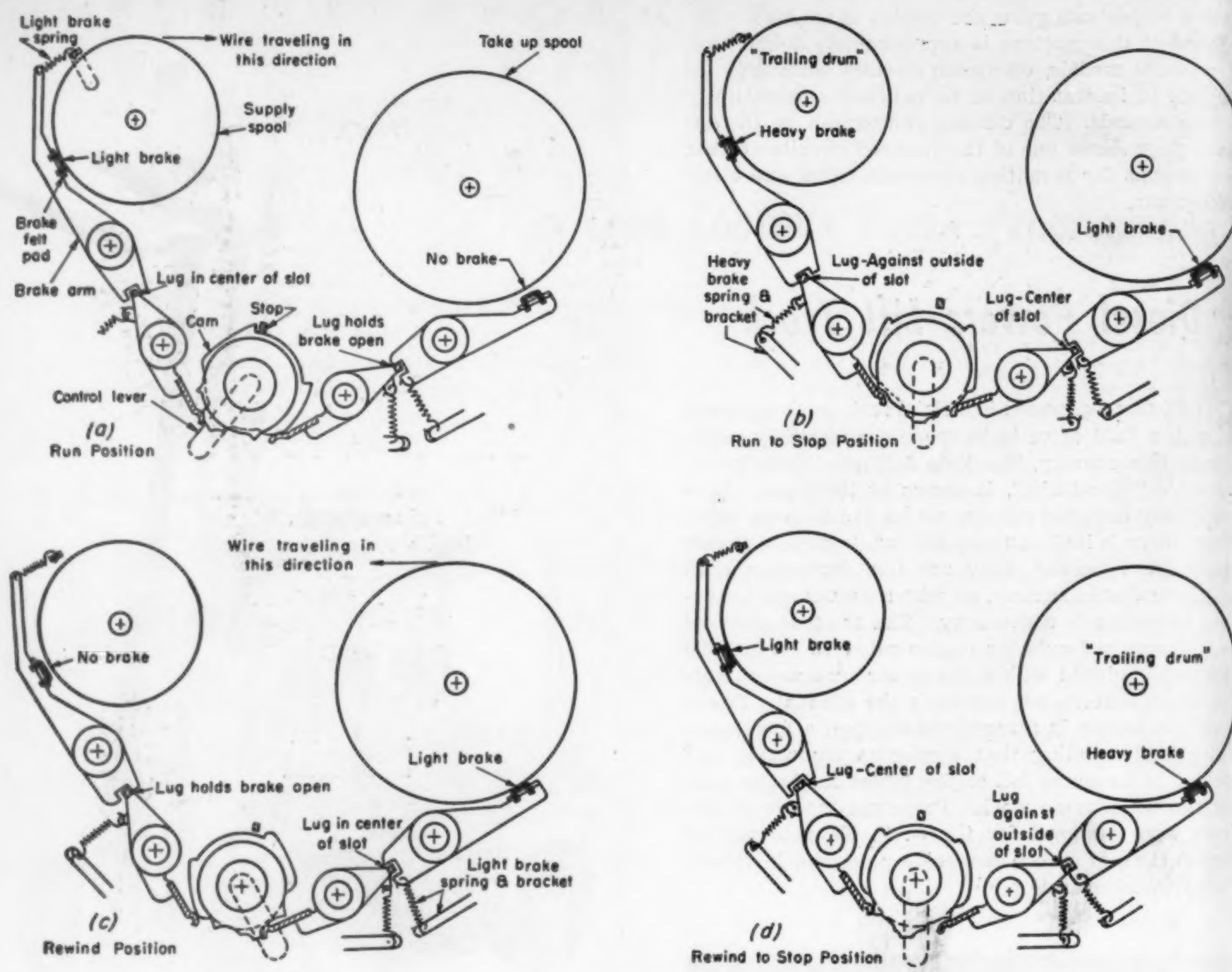
Speed of this machine is approximately double that of previous models, electronic circuits permitting 80 columns of information to be punched automatically in four seconds. The closeup photograph on the opposite page shows one of the stamped circuits utilized as a chassis for mounting electronic tubes and other components.

Diesel Powers Lift Truck

FIRST diesel-powered industrial fork truck equipped with a fluid drive to be made commercially available in this country, The Yale & Towne Manufacturing Co.'s "Diesel-Lift", is shown at the right. It is specifically designed for use where fire hazards exist, where there is limited fresh air (while diesel exhaust fumes are nauseous, they are less dangerous than gasoline exhaust fumes), or where continuous heavy-duty operation is a necessity. The truck is powered by a Hercules 6-cylinder engine rated 70 hp at 2200 rpm and equipped with a condenser type water muffler which screens out sparks in the exhaust. Power from the engine is transmitted through a double-impeller fluid coupling that eliminates chattering and stalling and assures full engine power for heavy pulling and steep ramp work. Prototype models of this truck were designed for the U. S. Navy to protect personnel from carbon monoxide poisoning in closed-compartment operation.

Four-Way Brake Controls Recorder

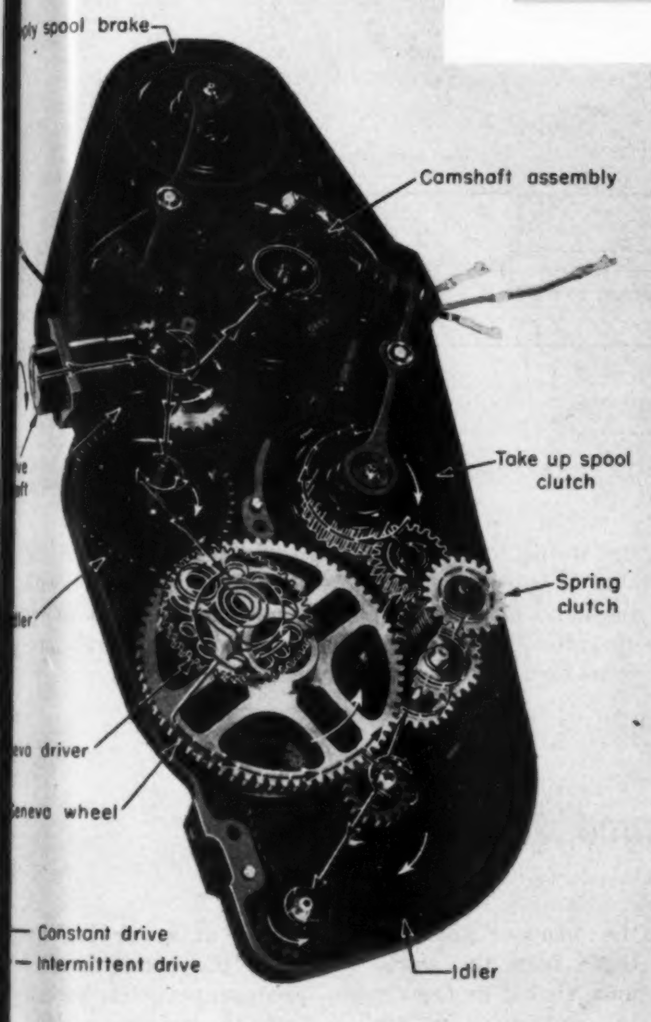
INGENIOUS braking system on the Webster-Chicago model 288 wire recorder, left, controls feeding of the wire by means of a "memory" device. Four different braking conditions are encountered in the recorder—one when recording, one when rewinding, one when stopping after recording, and one when stopping after rewinding. A system of pivoted spring-loaded brake levers, tied in with two cams on the control lever, operates the brakes as follows: When the control lever is moved to the run position, a "light" brake is applied to the supply spool to prevent the wire from leaving the spool faster than the



takeup spool can wind it, *a* in the drawing above. Returning the control lever to the stop position causes a heavy brake to be imposed on the supply chuck and, through the upper of two cams and the right-hand follower, a light brake is impressed on the takeup spool, as at *b*. Different brake conditions are imposed, *c*, during rewind operation. In shifting from rewind to stop, however, a different brake condition, *d*, is required than when shifting from run to stop, although the control lever is in the same position. This is accomplished by the "floating" upper cam which, because of the large backlash present in an oversize keyway, can shift relative to the lower cam. This upper cam is rotated to a different position when changing from run to stop than when shifting from rewind to stop, providing different braking for the two positions. Cam followers can assume three positions: centered in their slots in the brake levers to permit light braking; held against the outside of their slots (with follower tips floating on the cams) to permit heavy braking; or held against the inside of their slots, holding the brakes off.

Geneva Transmits Film in Camera

GENEVA drive in the magazine of the aerial camera, top right, next page, made by Ryan Industries Inc., provides the intermittent dwell necessary for rapid "still" photography. This magazine automatically presents the correct length of unexposed film (9 by 18-inch negative size) to the focal plane of the camera immediately following each exposure, at speeds of one exposure every 1.2 seconds. Mechanical actuation of the various film-control assemblies is provided through the magazine drive,

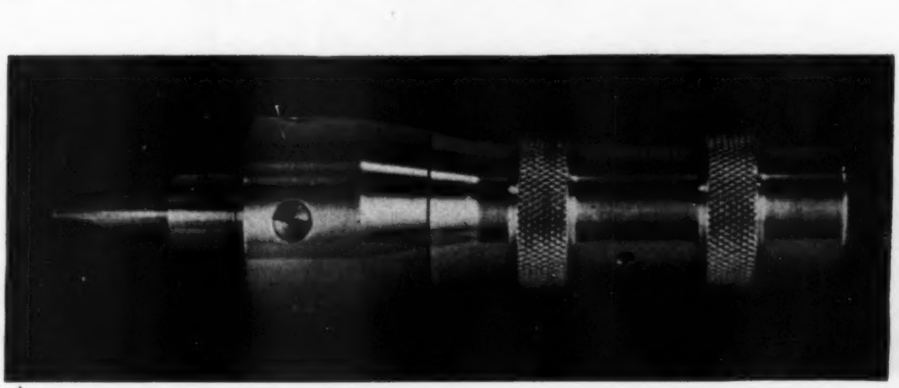


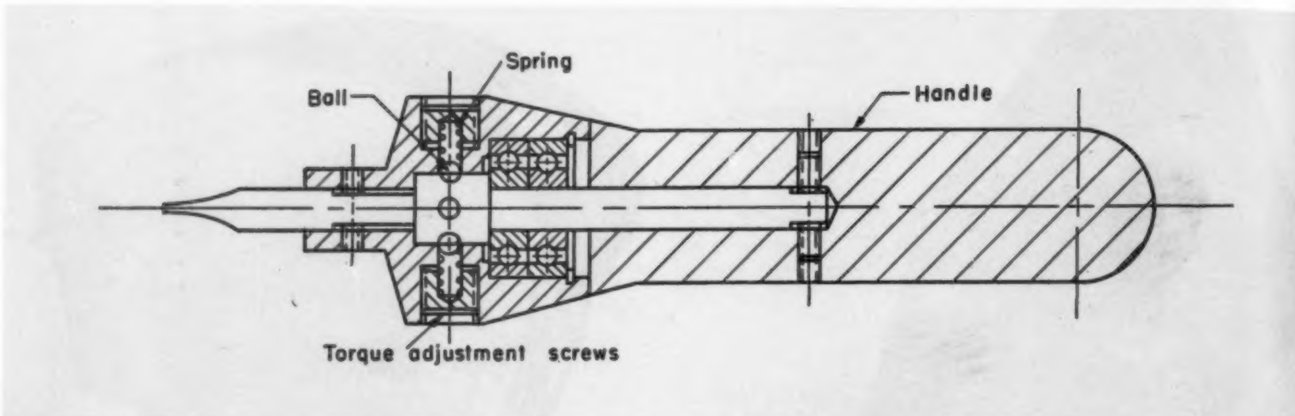
above. The input gear train is coupled to the camera case driveshaft through a linkage which rotates the main camshaft assembly and the Geneva. Operation of a vacuum solenoid switch and an operation indicator switch, as well as the clutch and supply spool brake, are controlled from the camshaft.

Geneva movement which transfers film from the supply spool to the takeup spool consists of a gear-driven input wheel which engages an output wheel geared to the film-drive mechanism to give the necessary intermittent film movement. Both the metering roll which pulls the film from the supply spool and the spool which winds up the film are driven by the Geneva gearing; their rotation, therefore, is also interrupted by the dwell period. Plastics are used in many of the magazine gears and other parts, permitting weight reduction and resulting in a quiet mechanism that operates at extreme temperatures.

Screwdriver Has Overload Release

MECHANICAL screwdriver for applying a specific preset torque — preventing stripped threads and damaged material — without friction clutch or other delicate adjustment devices is shown at the right. Made by Overload Control Co., the screwdriver employs a mechanism, drawing top of next page, that requires few

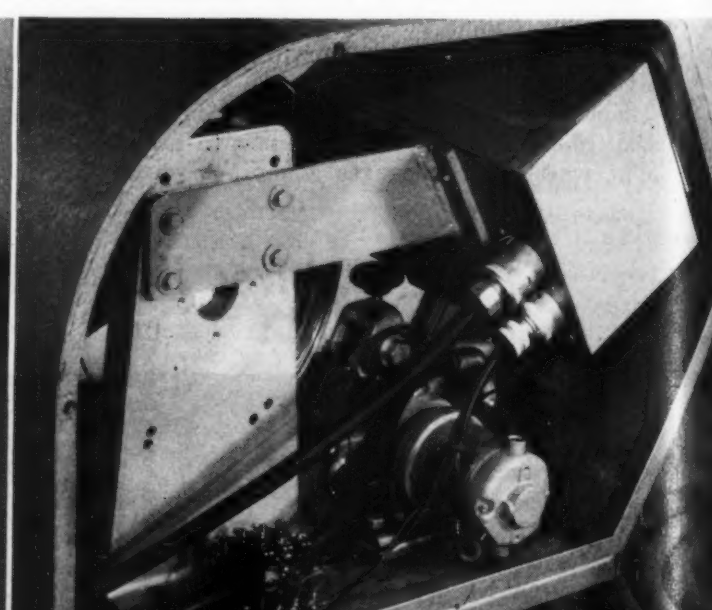




moving parts and is easily adjusted. Turning the four screws changes the spring load on steel balls which engage sockets or detents in the output shaft of the screwdriver. When the preset torque is exceeded, the balls are forced from their sockets against the spring forces and disengage the handle or driving end. Torques up to 15 inch-pounds are available with the hand model; another model, adaptable for power operation in a drill press or electric hand drill, provides torques to 25 inch-pounds.

Manual Welder Designed for Portability

EASY operation and low maintenance costs are obtained with the compact portable manual welder, below, because the low-voltage rod-feed motor and the series type motor voltage control have few parts and require minimum adjustment. Controls in the UWM-1 machine, made by Linde Air Products Co., automatically maintain constant welding voltage even though the operator holds the hand unit at a varying distance from the work. Parts of the caster-mounted unit visible in the closeup photograph below include the welding head with a 3-pound capacity welding composition hopper, a voltage-control box and a 75-pound capacity rod reel mounted in the small steel chassis and a 17-foot hose to the hand unit.



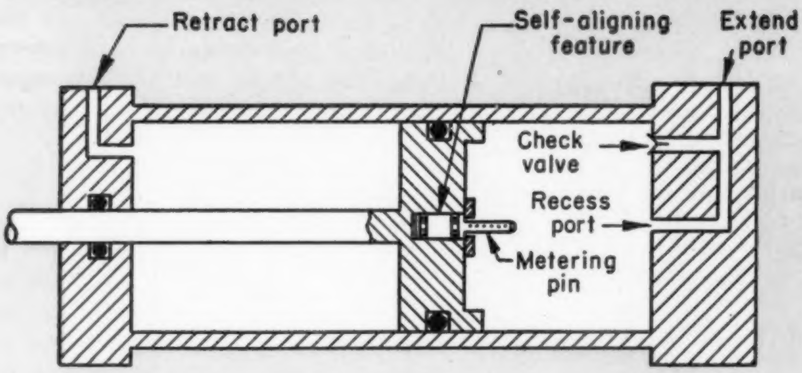


Fig. 1—Hydraulic cylinder with integral dashpot. Impact is absorbed at end of stroke by controlled flow through radial metering-pin orifices

Designing an Integral Dashpot

. . . for a hydraulic cylinder

By William E. Soong
Armour Research Foundation
Chicago, Ill.

A TAPERED pin attached to the piston of a hydraulic cylinder has often been considered for use in an aircraft hydraulic system to absorb an impact load at the end of the stroke. However, such application has not proved practical because of the difficulties involved in the fabrication of a design using a varying annular metering orifice formed with a tapered pin in a recess port. The main reason for the failure of such a design is that the amount of oil available for the metering purpose is usually limited. A small amount of oil would require a tapered pin with extremely precise taper and dimensions. In the design discussed in this article, *Fig.* 1, a straight hollow pin with radially drilled orifices is employed instead of the tapered pin. Test results obtained from the design closely approach the desired result for such a hydraulic unit.

Adaptation of an integral dashpot hydraulic cylinder gives the following advantages:

1. Automatic synchronization; no adjustments necessary
2. Integral construction; fabrication and installation costs substantially lower
3. Minimum wear; low maintenance cost
4. Operation relatively free from low temperature effects.

General Consideration: The hydraulic cylinder with integral dashpot can be utilized to advantage in such aircraft systems as wing fold, landing gear, canopy and arresting hook. An actuating cylinder with a properly designed integral dashpot permits speedier operation and prevents inertia loads from injuring the surrounding structures by introducing a new method of energy absorption. This is accomplished by provision of a gradually decreasing orifice area which is obtained from the design of the component parts of the integral dashpot. These parts can be divided for purposes of discussion as follows:

1. Pin with metering orifices
2. Self-aligning feature for the pin
3. Free flow device to assist the return stroke.

Of these, 2 and 3 present no special problem. Therefore, attention will be directed primarily to the design of a pin with metering orifices. The following fac-

Developments reported in this article were undertaken while the author was hydraulic analytical and design engineer with Chance Vought Aircraft Co.

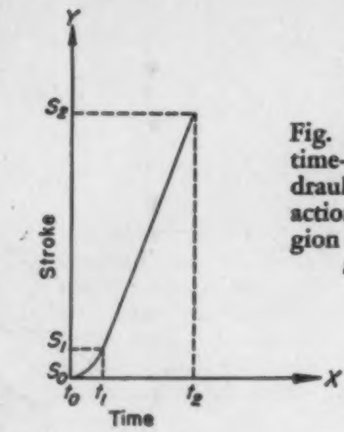


Fig. 2 — Left — Typical time-stroke curve for hydraulic cylinder. Dashpot action occurs within region bounded by points t_0, S_0 and t_1, S_1

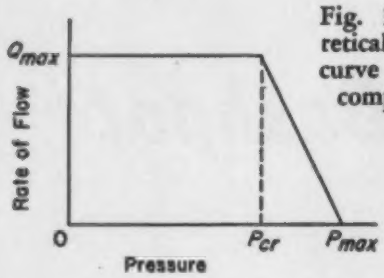


Fig. 3 — Left—Theoretical flow - pressure curve of a pressure-compensated pump

tors must be considered in this development:

1. Diameter of the pin
2. Size of the metering orifices
3. Number and location of the metering orifices.

THEORETICAL ANALYSIS: Piston diameter, piston stroke and pump characteristics are easily ascertainable. A dashpot stroke of one inch will suffice for most purposes.

Diameter of Pin: The metering pin should be hollow and the wall thickness should be a minimum. The outside diameter of the pin is arbitrarily chosen with consideration being given to stiffness and to manufacturing feasibility. In most cases, 3/8-inch to 1/2-inch diameter is reasonable. However, clearance between the pin and the recess port should be kept at a minimum to prevent any appreciable leakage of fluid. Yet, this minimum clearance must be sufficient to prevent any Brinelling or scoring of the pin and the recess port. From laboratory tests, it was found that a minimum clearance of 0.0015-inch and a maximum of 0.0025-inch give good results. Therefore, if the pin has a diameter of 0.3735, plus 0.0000 minus 0.0005, the recess port should have a diameter of 0.3750, plus 0.0005 minus 0.0000.

Size of Metering Orifices: Several considerations determine the size of the individual metering orifices. Theoretically, the greater the number of orifices, the better the flow distribution through the total orifice area resulting in a smoother dashpot action. However, it is impractical to use the smallest drill size available because of breakage.

From laboratory tests, it was found that drill sizes between No. 70 and 74 (0.028 to 0.0225-inch diameter) furnish a reasonable compromise between the practical requirements and the theoretical desirability of the maximum number of orifices.

Number and Location of Orifices: The first step is to determine the shape of the time-stroke curve. A typical time-stroke curve is shown in ***Fig.*** 2. The unmetered stroke and the corresponding operating time are as follows (see Nomenclature):

$$S_u = S_2 - S_1 \qquad (1)$$

$$t_u = t_2 - t_1 \qquad (2)$$

For $S_1 < S < S_2$, the velocity of the piston is

$$\frac{dS}{dt} = \frac{S_2 - S_1}{t_2 - t_1} \qquad (3)$$

The oil volume of the unmetered or free stroke is

$$V_u = A_p\ (S_2 - S_1) \qquad (4)$$

A theoretical flow characteristic curve of a typical pressure compensated pump is shown in ***Fig.*** 3. The velocity of the piston for $S_1 < S < S_2$ can also be expressed as

$$\frac{dS}{dt} = \frac{Q_{max}}{A_p} \qquad (5)$$

Nomenclature

$A_d = A_p - A_{pin}$ = Area of the piston during the dashpot stroke, sq in.
A_i = Individual orifice area, sq in.
A_o = Instantaneous orifice area, sq in.
A_p = Area of the piston, sq in.
A_{pin} = Area of the pin, sq in.
A_t = Total orifice area, sq in.
C = Coefficient of discharge through an orifice
D_p = Diameter of the piston, in.
D_{pin} = Diameter of the pin, in.
K = Increment designation (TABLE 2)
n = Exponent of the time-stroke exponential curve
N = Total number of orifices
P = Instantaneous pressure, psi
P_{cr} = Pump critical pressure, psi
P_l = Pressure due to mechanical load, psi
P_{max} = Maximum pressure of pump, psi
P_p = Pump pressure, psi
Q = Instantaneous rate of flow, cu in. per sec (or gpm)
Q_{av} = Average rate of flow, cu in. per sec (or gpm)
Q_{avp} = Average rate of flow of pump, cu in. per sec (or gpm)
Q_{max} = Maximum rate of flow of pump, cu in. per sec (or gpm)
S = Instantaneous stroke, in.
S_u = Total unmetered stroke, in.
S_0 = Zero stroke, in.
S_1 = Dashpot stroke, in.
S_2 = Cylinder total stroke, in.
t = Instantaneous time, sec
t_u = Operating time of the unmetered stroke, sec
t_0 = Zero time, sec
t_1 = Dashpot operating time, sec
t_2 = Cylinder total operating time, sec
V = Instantaneous volume of oil, cu in. (or gal)
V_u = Volume of unmetered oil, cu in. (or gal)
ρ = Mass density of the oil, lb-sec^2 per in.^{4}

For $S_1 > S > 0$, the theoretical curve which would produce the smoothest operation would have the same slope at the point (t_1, S_1) as that in Equation 3 for $S_1 < S < S_2$. Therefore,

$$\frac{dS_1}{dt_1} = \frac{S_2 - S_1}{t_2 - t_1} \qquad (6)$$

At the point (0,0), the curve should be tangent to the X axis or

$$\frac{dS_0}{dt_0} = 0 \qquad (7)$$

With these desired conditions, the mathematical representation of the curve during the dashpot stroke S may assume an exponential form such as

$$S = t^n \qquad (8)$$

Differentiating Equation 8 gives the velocity of the piston during the dashpot stroke.

$$\frac{dS}{dt} = nt^{n-1} \qquad (9)$$

Areas of the piston and the pin can be written as

$$A_p = \frac{\pi D_p^2}{4} \qquad (10)$$

$$A_{pin} = \frac{\pi D_{pin}^2}{4} \qquad (11)$$

Therefore, the piston area during the dashpot stroke is

$$A_d = A_p - A_{pin} = \frac{\pi}{4}(D_p^2 - D_{pin}^2) \qquad (12)$$

The total volume of oil in the dashpot can be written as

$$V_d = A_d S_1 \qquad (13)$$

Knowing the velocity of the piston, one can write the equation of the instantaneous rate of flow as

$$Q = A_d \frac{dS}{dt} \qquad (14)$$

Substituting Equation 9 into Equation 14,

$$Q = A_d n t^{n-1} \qquad (15)$$

Solving for t in term of S from Equation 8,

$$t = S^{\frac{1}{n}} \qquad (16)$$

By substitution of Equation 16 into Equation 15, the rate of flow becomes

$$Q = A_d n S^{\frac{n-1}{n}} \qquad (17)$$

Since the instantaneous rate of flow can also be written as the equation for flow through an orifice,

$$Q = CA_o \sqrt{\frac{2p}{\rho}} \qquad (18)$$

By equating Equations 17 and 18 and solving for A_o, the instantaneous orifice area is

Table 1—Orifice Area and Instantaneous Stroke

No.	A_s	S	No.	A_s	S
1	0.00641	1.000	9	0.00321	0.377
2	0.00601	0.912	10	0.00281	0.309
3	0.00561	0.827	11	0.00241	0.247
4	0.00521	0.744	12	0.00201	0.191
5	0.00481	0.663	13	0.00161	0.139
6	0.00441	0.586	14	0.00121	0.092
7	0.00401	0.511	15	0.00081	0.052
8	0.00361	0.441	16	0.00041	0.022

Table 2—Calculation of Time Increments

K	ΔS	Q	$A_s(\Delta S)$	Δt
1	0.088	30.43	0.803	0.0264
2	0.085	28.53	0.776	0.0272
3	0.083	26.63	0.758	0.0285
4	0.081	24.73	0.740	0.0299
5	0.077	22.84	0.703	0.0308
6	0.075	20.94	0.685	0.0327
7	0.070	19.04	0.639	0.0336
8	0.064	17.14	0.584	0.0341
9	0.068	15.24	0.621	0.0408
10	0.062	13.34	0.566	0.0424
11	0.056	11.44	0.511	0.0447
12	0.052	9.54	0.475	0.0498
13	0.047	7.64	0.429	0.0562
14	0.040	5.74	0.365	0.0636
15	0.030	3.85	0.274	0.0712
16	0.022	1.95	0.200	0.1026
Total	1.000		9.131	0.7145

$$A_o = \frac{A_d n}{C\sqrt{\frac{2p}{\rho}}} S^{\frac{n-1}{n}} \qquad (19)$$

Let A_t and A_i be the total orifice area and individual orifice area, respectively. Then by substitution of S_1 for S in Equation 19, the total orifice area required is

$$A_t = \frac{A_d n}{C\sqrt{\frac{2p}{\rho}}} S_1^{\frac{n-1}{n}} \qquad (20)$$

Then

$$N = \frac{A_t}{A_i} = \frac{A_d n}{CA_i\sqrt{\frac{2p}{\rho}}} S_1^{\frac{n-1}{n}} \qquad (21)$$

Equation 19 gives the dynamic relationship between the instantaneous orifice area and the instantaneous stroke. Therefore, Equation 19 and Equation 21 give the number and location of the orifices. In order to clarify this further, a typical example will be presented later.

The volume of the dashpot can be expressed as

$$V_d = \int dV \qquad (22)$$

And

$$Q = \frac{dV}{dt} \qquad (23)$$

By substituting Equation 23 into Equation 15, the

rate of flow is expressed by

$$\frac{dV}{dt} = A_d n t^{n-1} \quad \ldots\ldots (24)$$

By integrating Equation 24, the total volume of the dashpot is

$$V_d = \int dV = \int_0^{t_1} A_d n t^{n-1} dt$$

$$= A_d n \left.\frac{t^n}{n}\right]_0^{t_1} = A_d t_1^n \quad \ldots\ldots (25)$$

By substituting Equation 16 into Equation 25,

$$V_d = A_d (S^n)^{\frac{1}{n}} = A_d S_1 \quad \ldots\ldots (26)$$

Equation 26 is identical to Equation 13, and therefore checks the validity of this analysis.

EXAMPLE: Assume the following values: For the actuating cylinder, $S_1 = 1$ inch, $S_2 = 11$ inches, A_p

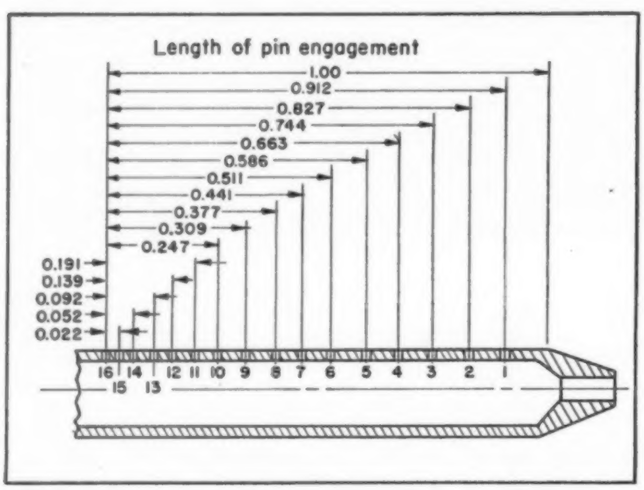


Fig. 4—Layout of orifice locations. Length dimensions for this example are developed in Tables 1 and 2

$= 9.24$ sq. in., $A_{pin} = 0.11$ sq in. For the pump, $P_{max} = 1000$ psi, $P_{cr} = 875$ psi, $Q_{max} = 9$ gpm $= 30.8$ cu in. per sec. For the fluid used (AN-VV-O-366b), $\rho = 0.000,079$ lb-sec² per in.⁴. For the orifice, $C = 0.77$.

From Equation 5, for $S_1 < S < S_2$, $dS/dt = Q_{max}/A_p = 30.8/9.24 = 10/3$ in. per sec. For *Fig.* 2, let $t_1 = 1$ sec, $t_2 = 4$ sec. From Equation 9 the slope at the point (t_1, S_1) is $dS_1/dt_1 = 10/3\ nt^{n-1} = n$. Substituting this into Equation 8 gives the equation of the time-stroke curve of the dashpot as: $S = t^n = t^{10/3}$. From Equation 12, $A_d = A_p - A_{pin} = 9.24 - 0.11 = 9.13$ sq in. Integrating Equation 15 and dividing by t_1 gives the average rate of flow throughout the dashpot stroke:

$$Q_{av} = \frac{1}{t_1}\int_0^{t_1} A_d n t^{n-1} dt = \frac{1}{t_1} A_d n \left.\frac{t^n}{n}\right]_0^{t_1} = \frac{1}{t_1} A_d t_1^n$$

$$= A_d = 9.13 \text{ cu in. per sec}$$

The average rate of flow during the dashpot stroke must also be the average rate of flow for the pump during the dashpot stroke. Therefore the average rate of flow of the pump during the dashpot stroke is: $Q_{avp} = 9.13$ cu in. per sec $= 2.37$ gpm. From *Fig.* 3, the corresponding pressure for the rate of flow of 2.37 gpm is, for $P_{cr} < P_p < P_{max}$:

$$P_p = \frac{-(P_{max} - P_{cr})}{Q_{max}} Q + P_{max} \quad \ldots\ldots (27)$$

$$= \frac{-(1000 - 875)}{8}(2.37) + 1000 = 963 \text{ psi}$$

Also any additional pressure acting on the fluid due to the mechanical load on the piston rod can be determined. In this example, the pressure due to mechanical load is assumed to be 537 psi. Therefore the total average pressure in the dashpot during the dashpot stroke is:

$$P = P_p + P_l \quad \ldots\ldots (28)$$

$$= 963 + 537 = 1500 \text{ psi}$$

The assumptions were made of a constant pressure in the dashpot and zero leakage through the annular orifice. The accuracy of the result based on these assumptions is well within the practical limits.

From Equation 20, the total orifice area required is

$$A_t = \frac{A_d n}{C\sqrt{\frac{2p}{\rho}}} S^{\frac{n-1}{n}} = \frac{9.13\ (10/3)}{0.77\sqrt{\frac{2(1500)}{0.000079}}}(1)^{\frac{(10/3)-1}{10/3}}$$

$$= 0.0064 \text{ sq in.}$$

In this example, No. 74 drill is used. The area is $A_i = \pi\ (0.0225)^2/4 = 0.000398$ sq in. From Equation 21, the number of orifices is $N = A_t/A_i = 0.00641/0.000398 = 16$. Equation 19 will give the location of each orifice if S is solved for by substitution of each incremental instantaneous orifice area for A_0 which, for convenience, is rounded to 0.00040 sq in. In this example, Equation 19 can be simplified as follows:

$$A_o = 0.00641\ S^{0.7} \quad \ldots\ldots (29)$$

Results of this calculation are shown in TABLE 1 which can be used for locating the orifices on the pin. *Fig.* 4 shows one method of distributing the orifices.

A time calculation for the design shown in *Fig.* 4 is desirable. Each time increment can be written as:

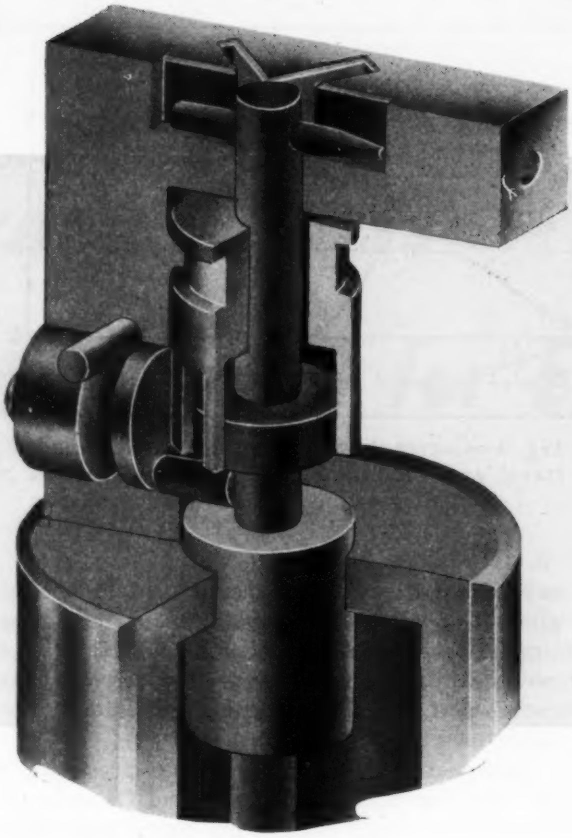
$$\Delta t_k = \frac{(\Delta S_k) A_d}{Q_k} \quad \ldots\ldots (30)$$

where

$$Q_k = CA_o\sqrt{\frac{2p}{\rho}} = 4747.4\ A_o \quad \ldots\ldots (31)$$

This calculation can be conveniently performed in tabular fashion, such as shown in TABLE 2. Total operating time from TABLE 2 is 0.7145-sec for the orifice distribution shown in *Fig.* 4. The difference of this total operating time from TABLE 2 and that of 1 sec, as assumed at the beginning of this sample calculation, has resulted from the particular incremental orifice areas utilized in this design. The operating time is not necessarily to be 1 sec as was assumed in the sample calculation. This difference of the calcu-

(*Concluded on Page* 198)



Complex Motion

. . . for selector mechanism obtained with only three moving parts

By E. W. Kuhn
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REQUIREMENTS of transformer tap-changer systems have led to the development of a simple mechanism giving combined intermittent rotary and reciprocating axial motions. In such systems tap-changer contacts must seat accurately in successive angular positions which are registered on an external indicator.

Previous designs used an operating handle or wheel located on the cover of the transformer. Recently, industry-wide standards have placed major importance on accessibility of the operating handle from areas around the transformer rather than from the cover. Usual designs for the transmission of motion under these conditions utilize a miter gear arrangement with a resulting rotary motion. Rotary motion only has reflected high assembly cost because of the small increment of rotation for each tap change and resultant difficulty in obtaining close alignment between the moving contact and the external position indicator. These conditions prompted the development of a more economical arrangement.

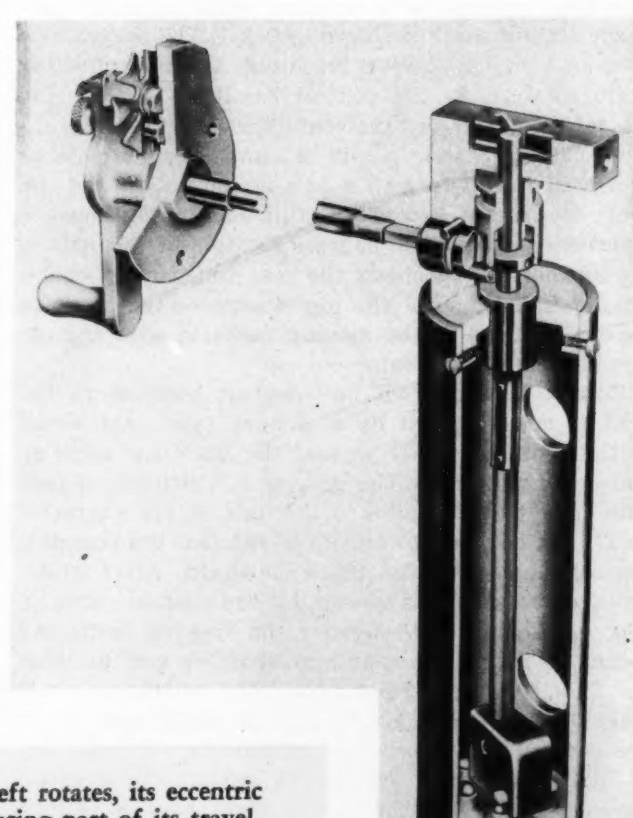


Fig. 1—Top—As the horizontal control shaft at the left rotates, its eccentric pin raises and lowers the vertical contactor shaft. During part of its travel, the pin also engages the Geneva type gear keyed to the contactor shaft and turns it to the next position

Fig. 2—Right—Only three moving parts are required to position accurately the contact mechanism shown in the lower part of the cylinder. Remote operation is controlled by the crank mechanism, at left, which includes a Geneva-gear arrangement for external indication

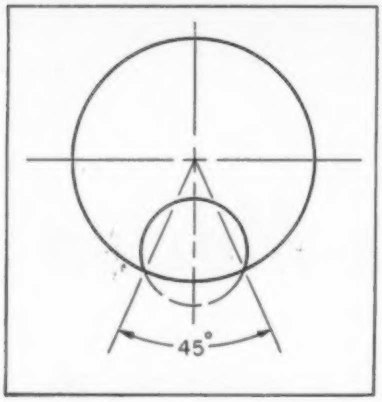


Fig. 3—The eccentric pin is cut away so that full downward pressure is maintained over 45 degrees rotation of the operating handle as shown in *Fig.* 4

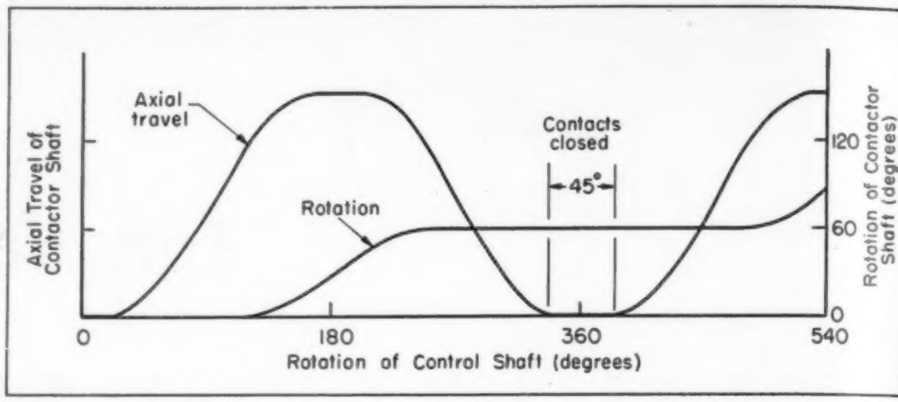


Fig. 4—Curves show the intermittent reciprocating axial travel and rotation of the contactor shaft in relation to rotation of the control shaft

Shown in *Fig.* 1, the new mechanism designed for this application is comprised of only three moving parts—an eccentric, a Geneva type gear and a special shaft. Sequence of operation for the new mechanism is to release contact pressure by raising the contactor shaft, rotate the shaft to the next position, and again apply contact pressure. Shown in partial section in *Fig.* 2 is the complete operating mechanism that was developed to meet these requirements.

A pin of special shape (*Fig.* 3) projects eccentrically from the end of a cylinder which is made part of the control shaft as shown in *Fig.* 1. The pin extends into an annular groove machined in the contactor shaft so that, as the control handle is turned, the pin raises and lowers the contactor shaft and its contacts. The eccentric pin is cut away on one side as shown in *Fig.* 3 to give a cam action so that as the shaft is pressed downward, full contact pressure is maintained through 45 degrees rotation of the operating handle. *Fig.* 4 shows the resulting motion curve. This special shape of the pin eliminates the need for close alignment of the moving contacts with the external position indicator.

Rotary motion from one contact position to the next is accomplished by a Geneva type gear keyed to the contactor shaft so that the shaft can slide up and down within it. The gear is cylindrically shaped with long teeth parallel to the axis of the contactor shaft. As the control handle is rotated, the eccentric pin begins to raise the contactor shaft. After 90 degrees of rotation, as shown by the motion curve in *Fig.* 4, the pin also engages the Geneva teeth and begins to rotate the contactor shaft as well as raise it. Shortly after 180 degrees, the contactor shaft starts downward while it still is being rotated by the gear. This combination of motions continues until the control shaft has turned through 270 degrees. Then the pin leaves the Geneva teeth, ceasing to turn the contactor shaft, and continues to depress the contacts until full pressure is established and the control shaft has completed a full revolution.

Accurate rotary positioning of the contactor is maintained by a pin set radially in the upper end of the contactor shaft. Radial grooves cast in the top of the tap-changer housing correspond with the six tap positions. As the contactor shaft is moved downward in any of these positions, the pin enters one of these grooves, aligning the contactor accurately and permitting no rotation. As the contactor shaft rises, the pin moves above the grooves where it can rotate to the next groove and drop into it as the shaft moves downward again.

Plaster-mold castings of the precision variety are used for both the housing and Geneva type gear with the resultant elimination of extensive machining operations. Axial constraint of the gear on the shaft without the use of assembly parts was accomplished by means of an integral shoulder within the housing and a matching groove in the gear. The control handle assembly is connected to the internal mechanism by a flexible shaft and slip joint with the square shank of the eccentric. Corresponding position indication between external and internal mechanisms is maintained by a small Geneva gear in the control handle assembly which is advanced one position with each revolution.

A compression spring retained within the molded contact arm assembly provides the required pressure on the contacts. Although no appreciable lubrication can be expected from the transformer insulating medium, wear on bearing surfaces is slight. Operation was normal after extensive life tests which simulated more severe conditions than would be expected during the normal service life for this class of apparatus.

They Say . . .

"The famous and much disputed analogy of the computing machine and the human brain is often misinterpreted as meaning that the machine possesses all the essential attributes of the brain. This is obviously absurd, since it could then, for example, design itself, but none the less, the analogy is very illuminating and suggestive if we recognize its limitations. The correct analogy is that between the brain and a *single run* on a computing machine, as the brain does not ever "clear" its memory completely during its existence as the machine does after finishing its task."—*The Engineers' Digest*, Jan. 1951.

PRODUCTION AND DESIGN

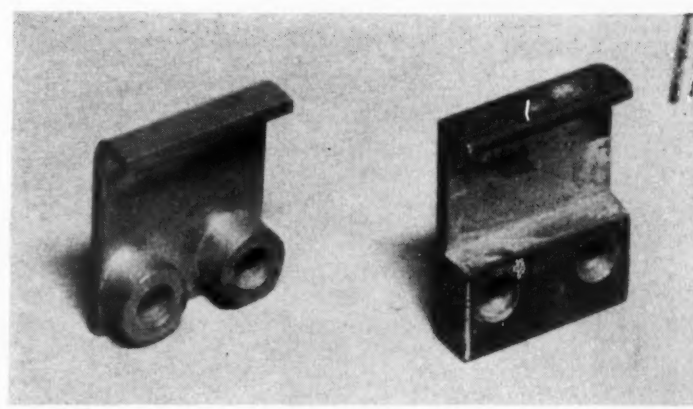
Modern Practices in Manufacture

Tractor Parts Redesigned for Economy

MORE and more attention is being focused in the direction of designing to suit the most practical method of production available. Today this means not only lower costs in dollars and cents but also fewer man-hours of labor and, most important, real savings in valuable materials. Typical of these efforts in the direction of better design are these following examples based on work presently being done by the manufacturing development division of the Caterpillar Tractor Co. of Peoria, Illinois.

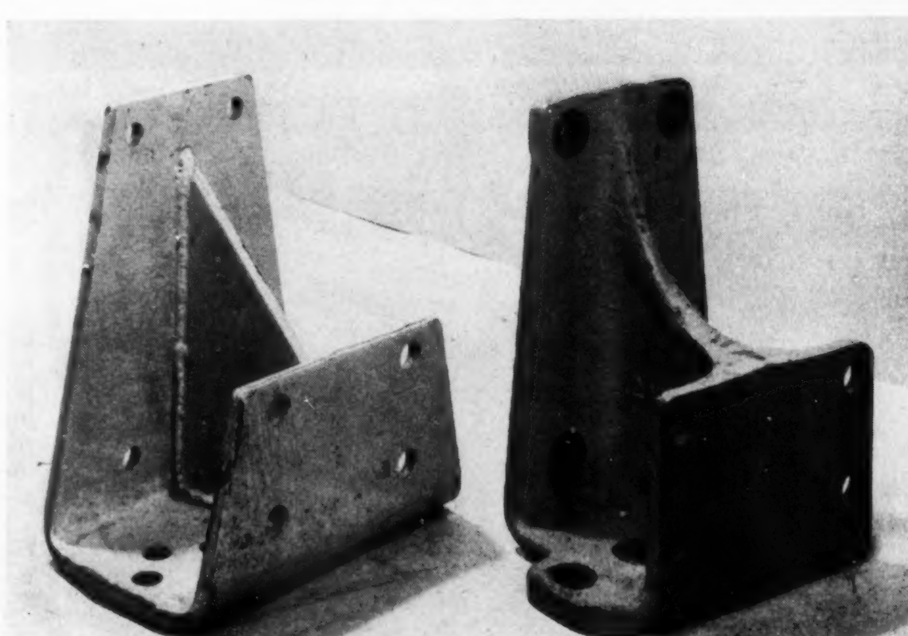
Governor Retainer

By replacing the steel machine part shown at the right with the sheet metal form and screw machine part assembly, a 20 per cent cost reduction was realized. The new assembly requires only press forming the sheet metal and force fitting the screw machine parts into the stamped holes. Expensive milling operations and considerable waste metal have been eliminated.



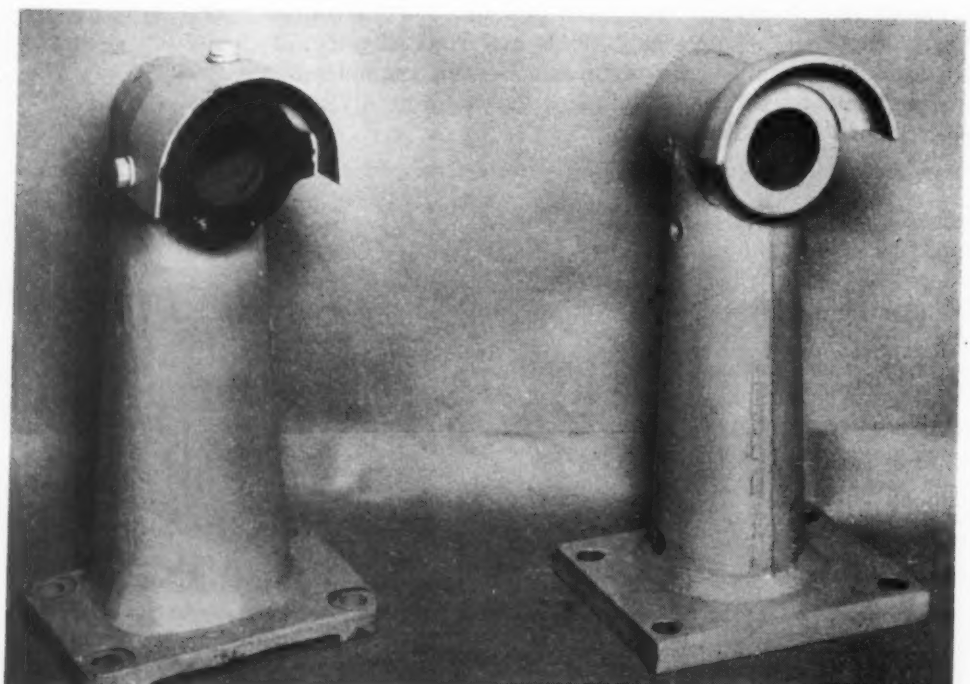
Support Arm

The formed and welded support at the right replaced the steel casting shown at far right. An overall reduction of 37 per cent in cost, and saving of three pounds in material resulted. Machining operations eliminated by the redesign included milling the top, bottom and end; spotfacing six bolt holes; and backfacing six bolt holes.



Support Bracket

A change in design from a steel casting, left, to a weldment, far left, is shown here. Redesigning this piece resulted in a cost reduction of 27 per cent largely from the elimination of two machining operations: (1) Milling the back face, and (2) spotfacing five holes. Tolerances on the angle section made finish machining unnecessary.



Carrier Bracket

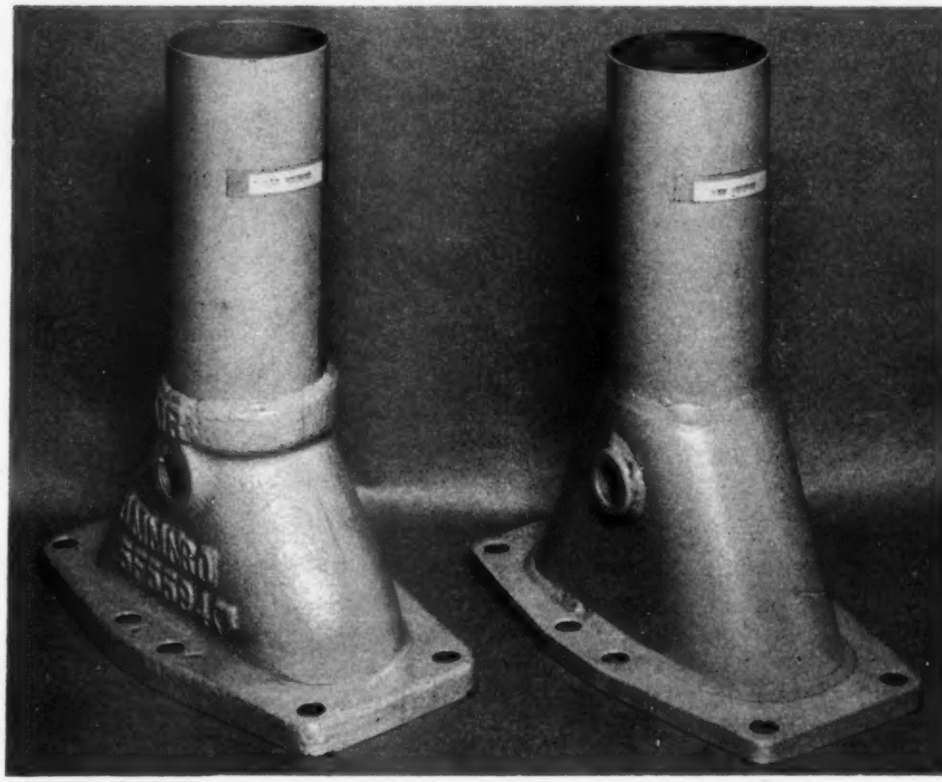
The redesigned bracket at the left is fabricated from mild steel plate and bar stock. It replaces the steel casting with attached guard at the left, resulting in an overall cost reduction of 32 per cent. A five pound material saving, reduction in assembly time, elimination of several machining operations as well as of three capscrews and lockwashers resulted from this new design.

Spring Housing

The spring housing design at the left—resulted in a 15 per cent cost reduction and a considerable saving in production time. This one-piece stamping replaced the brazed assembly at the left. The change was from a screw machine and blanked part assembly to a simple stamped unit requiring only a blank and draw operation. Elimination of the screw machine part required no substitute material and actually lighter gage sheet is used for the drawn design

Exhaust Stack

This exhaust stack assembly, right, was redesigned to reduce weight and cost. The casting in the design shown at the left was eliminated. The new design consists of a stamped base with stamped and formed reducer section welded together. The threaded side opening is replaced by a welded-in screw machine part. Milling the base also was eliminated and, in all, a 25 per cent cost reduction resulted.



Nut Guard

The function of this guard, below, is to protect exposed nuts or bolt heads on the under side of large track-type tractors. By replacing the forging at left with the press formed plate of heavy gage, a cost reduction of 20 per cent was realized. The simpler design effects the same protection.

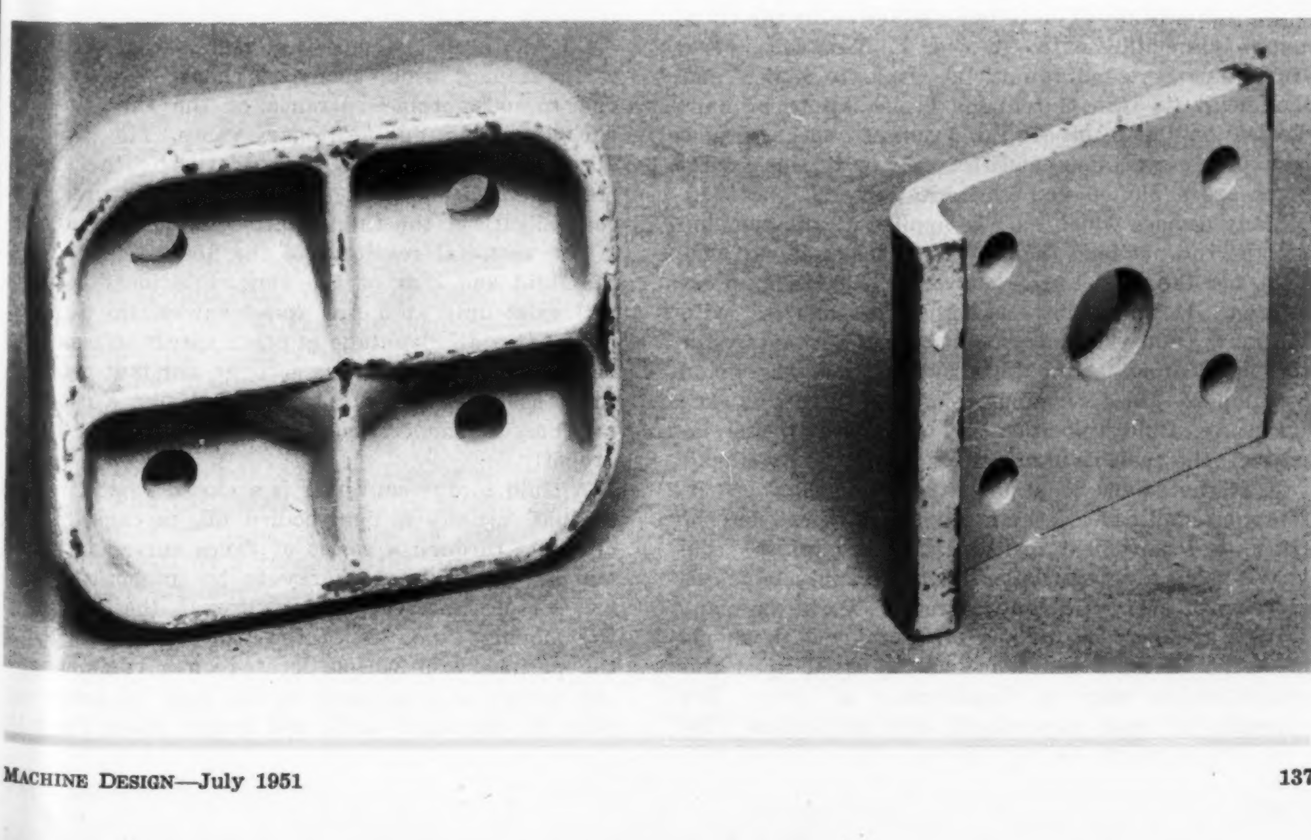
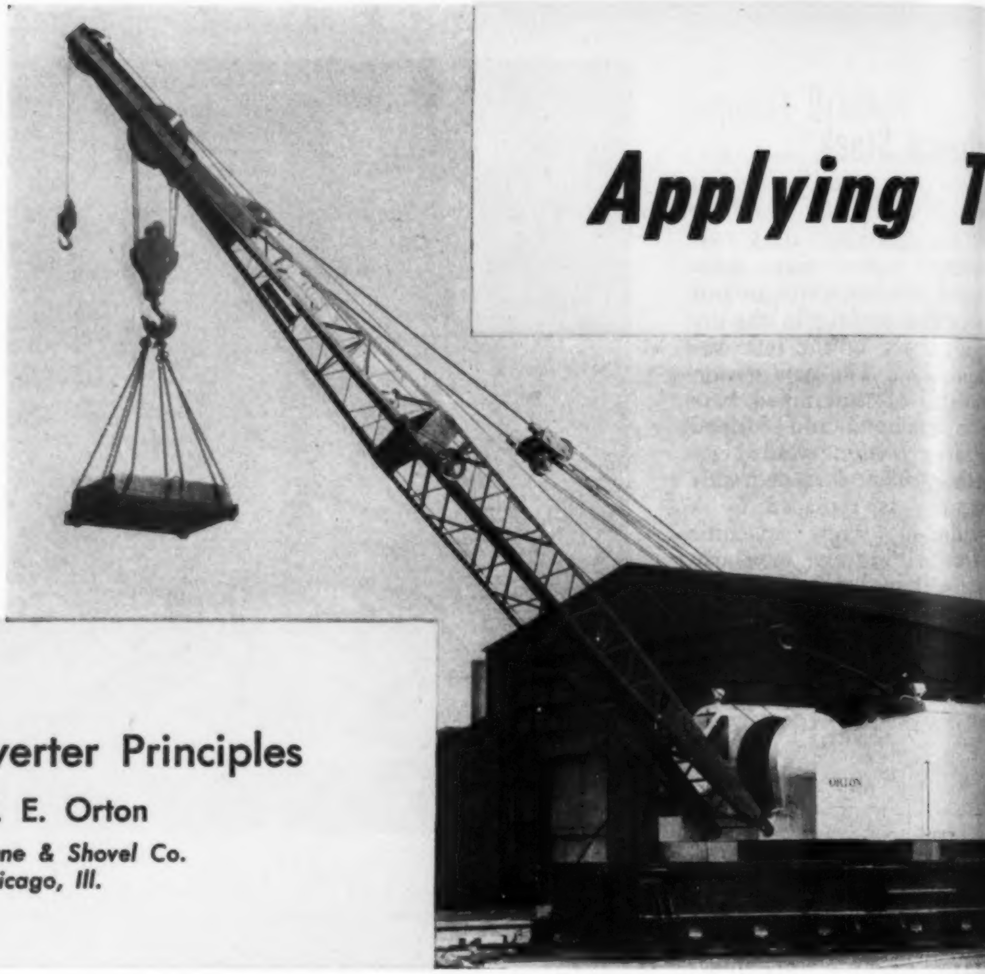


Fig. 7 — Heavy-duty twelve - wheel crane utilizes a torque converter drive



Applying T

Part 2–Converter Principles

By R. E. Orton

Orton Crane & Shovel Co.
Chicago, Ill.

A DEVICE is better applied if functionally understood. In particular is this true of the application of a torque converter to the locomotive crane illustrated in *Fig.* 7. This locomotive crane has four main motions, each in duplex and each with its own clutch and brake, all to be handled by one operator. The addition of another device in a power system is justified only when its advantages are exploited and its limitations not exceeded.

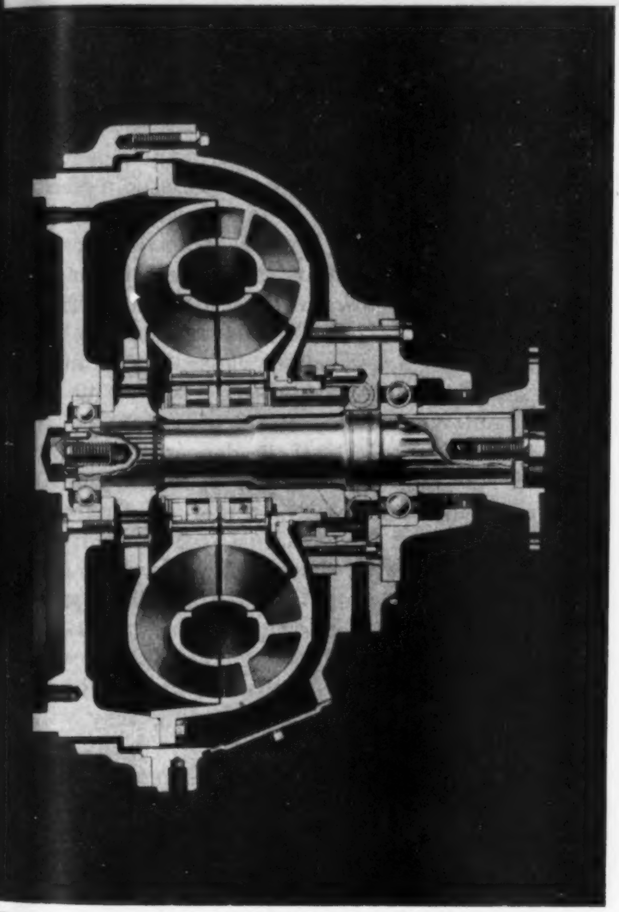
Many devices which may be applied by the machine designer do not lend themselves to a simple explanation, and the fluid torque converter, *Fig.* 8, is no exception. However, by taking some liberties with theory the converter may be discussed adequately for this application. Although a detailed analysis would be complex—the determination of such factors as vane curvature, fluid velocities and fluid drag being involved—the basic fluid principles are relatively simple.

It is customary to assume that fluid flow is perfectly streamlined. Nothing could be farther from the truth, the movement of the fluid through the converter being highly turbulent, with an appreciable dissipation of energy in eddy currents. While too many factors are involved for any other analysis, it should be borne in mind that conclusions based upon this simplification may be appreciably in error. For example, if true streamlined and frictionless flow existed, there would be no energy lost to the fluid with a stalled tail shaft whereas, actually all of the energy is lost to the fluid, the efficiency being zero.

One of the many important factors that contribute to turbulence, which increases energy losses, is that due to unfavorable entrance of the fluid into the various moving and stationary vanes. The ideal condition would be for the flow relative to the vane to be tangential as it enters the vane curvature. Since the velocity of the fluid relative to the moving vane is the vectorial resultant of the absolute velocity of the fluid and that of the vane, this ideal condition will exist only at a few speed values, the fluid entering at a disadvantage at other speeds. This cannot be taken into quantitative account, nor is it necessary to do so, as we are not trying to design a converter but only to understand it that it may be applied properly.

A fluid torque converter is a closed device wherein a fluid, usually a light-bodied oil, is caused to recirculate through a series of vanes curved in such a manner as to cause energy to be transmitted from one set of vanes to another through the medium of the moving fluid. Since the vanes are attached to the periphery of wheels located on a common axis, the energy is manifested in the form of torque applied to the input shaft and torque taken from the

TUE CONVERTERS to Crane Drives



Illustration, courtesy Detroit Diesel Engine Division, General Motors Corp.

Fig. 8—Sectional view of a torque converter showing impeller designed for mounting on engine flywheel, turbine splined on take-off shaft, and two stator elements mounted on one-way clutches for free wheeling

output shaft. By fixing a third set of vanes to form a stator the torque from the input wheel may be multiplied (converted) at the tail shaft.

VANE THEORY: Starting with the basic element—a single vane over which a stream of fluid is flowing, as illustrated in *Fig.* 9—consider the vane stationary with the fluid entering tangential to the blade curvature. Since, in the ideal concept, there is no friction loss in the fluid the (scalar) velocity at exit is the same as that at entrance. A thrust or push on the vane is developed as a result of the force required to change the direction of flow, as evidenced by the fundamental law of motion wherein force equals mass times acceleration. The acceleration is the change in the direction of the fluid velocity induced by the vane, the fluid entering at angle α and leaving at angle β as shown in *Fig.* 9.

A useful quantitative relationship for the force on the vane may be developed from this Law which is expressed as

$$F = Ma \qquad (1)$$

where F is the force of reaction, in pounds, M the mass expressed in mass units ($M = W/g$ where W is the weight in pounds and g the acceleration of gravity) and a the acceleration in the direction of F.

Reaction to Slug of Fluid

Considering a slug of fluid as shown in *Fig.* 9, of cross-section area u and length l, the mass of the slug is $\rho u l$, where ρ is the fluid density in units of mass per unit volume. The total vectorial change in velocity of the slug in the direction of R in passing across the vane, is ΔV, as shown in the vector diagram of *Fig.* 9. The time, Δt during which ΔV occurs is the time the slug is on the vane, which will be L/V seconds (V being the scalar value of the fluid velocity across the vane). Assuming that a may be represented by the average acceleration $\Delta V/\Delta t$, the reaction ΔR set up by the slug is given by

$$\Delta R = (\rho ul)\left(\frac{\Delta V}{\Delta t}\right) = (\rho ul)\left(\frac{V}{L}\right)(\Delta V)$$

With a continuous flow process such as this, there will be L/l slugs of fluid on the vane at all times, so that

$$R = \Sigma(\Delta R) = \left(\frac{L}{l}\right)(\Delta R) = \rho uV(\Delta V)$$

The volume Q of fluid flowing over the vane, per second, is uV and, substituting,

$$R = \rho Q(\Delta V) \qquad (2)$$

That is, the reaction force of the fluid on the vane is equal to the product of the fluid density, the volume of flow, and the vectorial change in velocity in the direction of the reaction.

This relation is developed in terms of Q because, inasmuch as the converter furnishes a closed circuit, at any instant the quantity of flow at any section of the converter will be the same as at any other section. Therefore, at any instant, reaction on the vane is dependent solely on the change in velocity.

The general case, a vane moving with velocity S

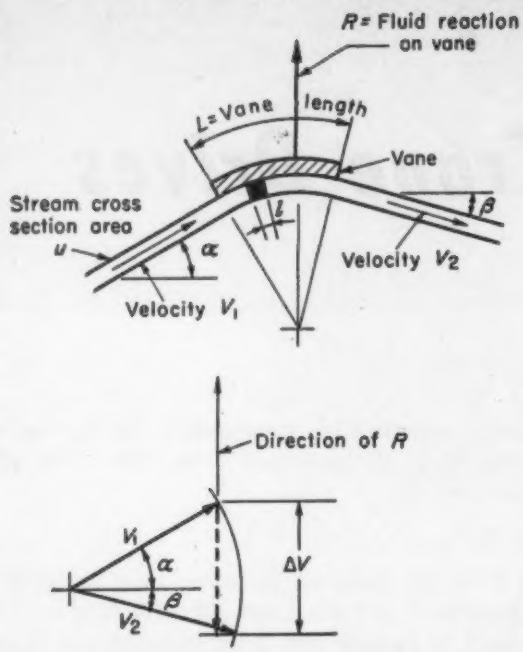
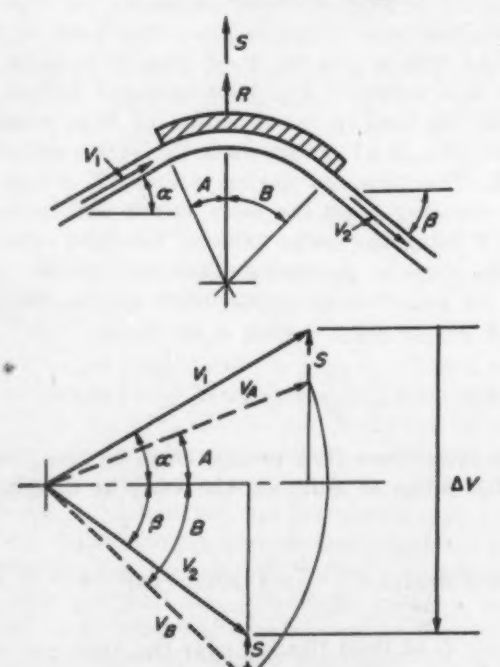


Fig. 9—Above—Fluid reaction on a stationary vane and velocity diagram showing reaction

Fig. 10—Below—Fluid reaction on a moving vane and vector diagram



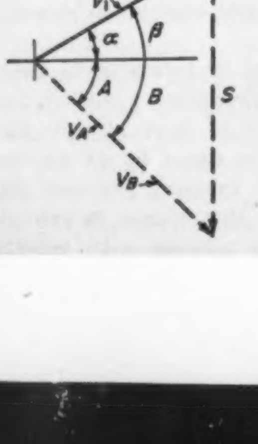


Fig. 11—Left—Vector diagram for vane moving at synchronous speed

in the direction of R, is shown in ***Fig.*** 10. The fluid stream approaches with velocity V_1 at angle α. Since the vane is moving with velocity S, the fluid velocity relative to the vane is the vectorial difference between V_1 and S, shown in the diagram as V_A and angle A. Since it is assumed that there is no friction loss along the vane, the relative velocity at exit is changed only in direction, to V_B at angle B. The absolute velocity at exit is then the vectorial sum of V_B and S, or V_2 (at angle β) as shown in the diagram. Letting ΔV, as before, represent the vectorial difference of V_1 and V_2, equation 2 will still apply.

Fluid Acts Equally on all Vanes

In the application of Equation 2 to the general case illustrated by ***Fig.*** 10, ρ may be considered a constant. While Q will vary, as previously mentioned, it has the same value at every section at any moment and, therefore, any variation of its value will have the same effect on every vane along the fluid traverse. Then Q may be treated, for the moment, as a constant considering its variation separately as an overall effect acting on all vane sets alike. This leaves ΔV as the only variable.

Unfortunately, ΔV itself is dependent upon four variables—the approach velocity V_1, angle α, the vane velocity S, and the relative angle of departure B. Since it is assumed that the fluid will leave with relative velocity tangential to the vane curvature at exit, and since this curvature is fixed by the design, B may be treated as a constant. V_1 and α will vary widely, being dependent upon the action of preceding vanes and other factors. However, in order to study the effect of varying vane velocity S, they will for the moment be treated as constants.

Referring again to ***Fig.*** 10, if $S = 0$, angle $A = \alpha$, $B = \beta$, giving the stationary vane of ***Fig.*** 9. Here ΔV, and therefore R, has its highest value (for positive values of S). As S increases in value ΔV decreases and, therefore, the thrust R decreases. Angle A decreases to zero and becomes negative until, at some value of S, angle $A = -B$, as shown in ***Fig.*** 11. Here V_A coincides with V_B, as does V_2 with V_1, ΔV being zero. The vane is moving with the stream, no thrust being developed.

Another useful concept may be developed by consideration of the energy delivered by a stream, the kinetic energy being

$$E = \tfrac{1}{2}MV^2$$

For a slug of fluid, $M = \rho ul$ and its kinetic energy is

$$\Delta E = \tfrac{1}{2}\rho ulV^2$$

The number of slugs delivered per second is V/l, so that

$$E = \Sigma(\Delta E) = \left(\frac{V}{l}\right)(\Delta E) = \tfrac{1}{2}\rho uV^3$$

gives the energy delivered per second. As before, the volume delivered is $Q = uV$ and, substituting

$$E = \tfrac{1}{2}\rho QV^2 \quad \text{(3)}$$

Referring to *Fig.* 10, the energy brought by the stream is $\frac{1}{2} \rho Q V_1^2$, and that left in the stream on exit from the vane is $\frac{1}{2} \rho Q V_2^2$. The difference in these two represents the energy delivered to the vane, which will also be equal to the work per second done by the vane, that is, RS. Therefore

$$\tfrac{1}{2}\rho Q V_1^2 - \tfrac{1}{2}\rho Q V_2^2 = RS$$

from which

$$V_2^2 = V_1^2 - \frac{2RS}{\rho Q}$$

and, substituting for R from Equation 2,

$$V_2^2 = V_1^2 - 2S(\Delta V) \quad \text{(4)}$$

Also, from the foregoing, the work (or energy) delivered to the vane per second is RS and, substituting for R,

$$W = \rho Q S(\Delta V) \quad \text{(5)}$$

In a converter the stator vanes are fixed vanes. For this condition $S = 0$ in *Fig.* 10, giving the condition of *Fig.* 9. From Equation 4 $V_2 = V_1$ and from Equation 5 work is zero. No energy is taken from the stream, a stator serving only to change the direction of flow.

If S has a positive value, as shown in *Fig.* 10, V_2 is less than V_1 and work is positive, energy being taken from the stream. This is the condition of a turbine vane. If $S = 0$ is now considered as the limit condition, it may be treated as a stalled turbine vane. As S increases from zero ΔV (and R) decrease, V_2 decreases to some minimum value at which point the energy received by the vane, Equation 5, reaches a peak. Further increase in S increases V_2 until the condition of *Fig.* 11 is reached where V_2 matches V_1, ΔV and therefore R and the work are zero. Here the turbine vane is running at what might be called the synchronous speed.

Supplies Energy to the Fluid

If S is increased beyond this point, ΔV is negative, as shown in *Fig.* 12, V_2 becomes greater than V_1, and energy is added to the stream. R is reversed in direction so that a force is required to drive the vane (furnishing the energy added to the stream). This gives a "braking" condition for the turbine vane.

Returning to the $S = 0$ condition and then reversing the direction of S so that it has a negative value as shown in *Fig.* 13, ΔV remains positive, and, from Equations 4 and 5, V_2 is greater than V_1 and energy is added to the fluid. Since the vane is moving against R it must be driven, thus supplying the energy added to the fluid. This gives the condition of an impeller vane.

This discussion shows that whether a vane is an impeller, a stator or a turbine is purely a matter of how it is employed. There is no theoretical difference. In an actual converter there are a series of vanes arrayed around the periphery of each wheel or rotor. The fluid also fills all the space between the vanes. If the difference in radial position of the vane elements is ignored, the linear relationships just de-

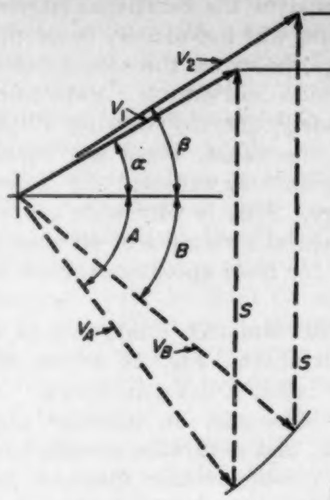


Fig. 12—Left—Vane moving faster than synchronous speed

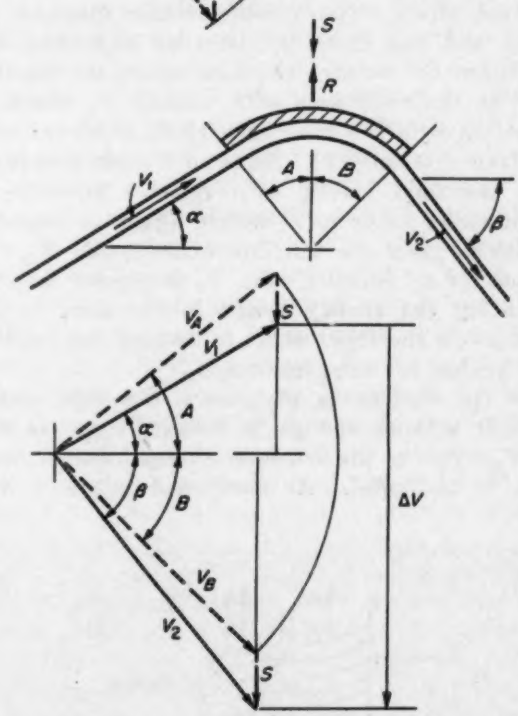
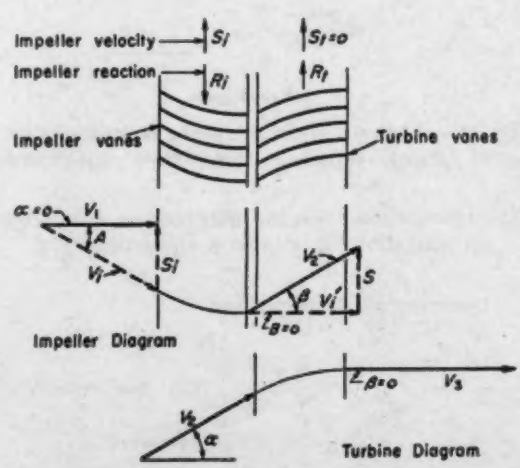


Fig. 13—Above—Diagram for vane when moving in reverse

Fig. 14—Below—Conditions for stalled fluid coupling, showing impeller and turbine diagrams with fluid entering tangentially



veloped may be used to analyze the rotational effects.

In the converter the fluid will not usually enter the vanes tangentially, as here indicated, the actual vanes having a fixed compromise curvature. Vane and fluid velocity varying widely, thereby varying velocity direction relative to the vanes, there are bound to be entrance losses which will substantially affect the converter performance. This is why wide torque range is inevitably obtained at a sacrifice of efficiency, and why a unit designed for fixed speeds can show so much less lost energy.

Fluid Couplings: The simplest fluid unit is a coupling with stalled tail shaft. *Fig.* 14 shows the vanes in what might be called a developed view. A coupling has two rotor elements, an impeller connected to the input shaft, and a turbine mounted on the output shaft. The velocity vector diagram for entrance and exit from the impeller and turbine is shown below the vanes. The fluid enters the impeller parallel to the rotor axis with velocity V_1 which, in combination with the vane velocity S_i, gives the relative entrance velocity V_i. Since no friction loss is assumed, the fluid leaves with relative velocity V_i' equal in scaler value to V_i which, again, in combination with S_i gives the absolute exit velocity V_2. As discussed for an impeller vane, V_2 is greater than V_1, representing the energy stored in the fluid by the work done on the input shaft in forcing the impeller around against the vane reaction R_i.

Since the turbine is stationary the fluid passes through it without change in velocity value so that $V_3 = V_2$. Due to the direction change, though, reaction R_t is developed. As mentioned before, a fluid coupling functions in a closed circuit, the vanes being so formed as to return the fluid discharged from the turbine directly into the impeller vanes. Therefore, V_3 must be equal to V_1 both in amount and direction. But it has been shown that $V_3 = V_2$ and that V_2 is greater than V_1. The error lies in ignoring the fluid energy losses which, under stall conditions, represent all of the input energy. This energy is dissipated in fluid friction, entrance and exit losses from the vanes, etc. In *Fig.* 14 the fluid is shown entering the vanes tangentially. Since a coupling is not usually designed for stall conditions this is not typical, entering conditions in an actual unit entailing considerable shock and turbulence.

Because of the short and free fluid path in the usual coupling design, energy balance with a locked turbine will occur at a high velocity of flow, giving high energy transference. High torque values will be required to maintain full input speed, in the order of ten or more times full-load rated torque in the usual design, so that in actual application the speed of the prime mover is pulled down to levels below full rated speed.

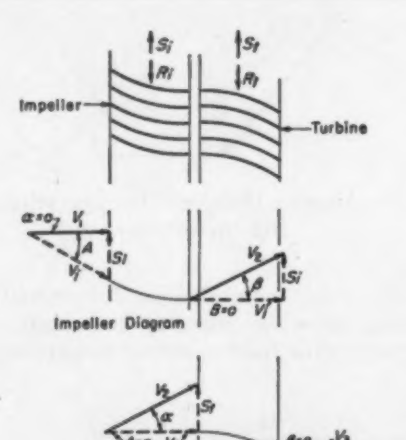
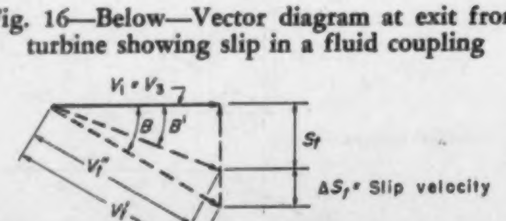


Fig. 15—Above—Fluid coupling running at impeller speed, showing fluid flow conditions

Fig. 16—Below—Vector diagram at exit from turbine showing slip in a fluid coupling



Torque Conversion Requires Stationary Vanes

Equation 2 shows that $R = \rho Q (\Delta V)$. Since the fluid leaves the turbine with the same vectorial velocity as it enters the impeller, it must be that whatever the velocity change ΔV in the impeller, it must be equal in value and opposite in direction in the turbine. Therefore $R_i = R_t$. In a coupling, then, output torque equals input. Fundamentally, then, in order to get torque conversion there must be a third set of vanes and they must be secured to a third shaft, or held stationary in some way. Torque conversion with only two elements is not possible.

Considering the fluid coupling at the opposite extreme to stalled, the turbine runs at impeller speed as illustrated by *Fig.* 15. The velocity vector analysis is apparent from inspection of the figure. In this case the turbine diagram is the exact reverse of that for the impeller. The vector analysis shows that the fluid is returned to the impeller at the same velocity as that at which it entered it, so that, if the fluid is set in motion at any given velocity, that velocity will be maintained. Again actual performance is modified by friction of flow so that V_i' at discharge from the impeller is somewhat less than V_i at entrance; and still further reduced at exit from the turbine. Since, as before, V_3 at exit from the turbine must be maintained equal to V_1 at entrance to the impeller, the energy must be obtained at the expense of turbine velocity. *Fig.* 16 shows this vectorially for the turbine discharge. V_t' is the theoretical discharge velocity which is reduced by friction to V_t''. Since the horizontal projection must equal V_3, S_t is reduced in value, V_t' changing in direction as well as quantity. A similar action (not shown) occurs at exit from the impeller, except that impeller velocity is maintained, V_2 being reduced in value and increased in angle. The reduction in S_t, denoted in *Fig.* 16 as ΔS_t, is known as the slip and is usually given in per cent of impeller speed. The corresponding reduction in the output energy from that of the input is the energy lost

in fluid flow. A coupling operating near input speed can be designed for excellent flow conditions so that slip at full torque and speed can readily be held to 3 per cent or less.

The force (torque) transmitted by the coupling is $R = R_i = R_t$. Referring to the vector diagram of *Fig.* 15, approximately, $\Delta V = S_i$ so that, for constant impeller speed, the torque transmitted is proportionate to the flow, Q, which in turn varies as the average fluid velocity, V. Substituting this proportionality in Equation 2 gives

$$R = kV \qquad (6)$$

where k is a constant.

Approximately, the friction head is proportionate to V^2 so that the lost velocity, δV (*Fig.* 16), is proportionate to V, and to ΔS, the slip. Therefore, for constant impeller speed the torque transmitted by a fluid coupling is proportionate to the slip.

Tests show that, at or near designed full load, this relation between torque and slip is quite close up to four or more times rated load. At that point, due to extreme turbulence and increasingly unfavorable vane angles, the flow resistance increases rapidly until a breakdown torque is reached, or the coupling stalls. This straight line torque relation means that a coupling operated near full design speed is not a "soft" energy transmitter, as it is usually thought, but relatively "hard" so that shock loads may draw from the inertia system of the prime mover.

The preceding may be generalized for varying impeller speed and slip. The discussion demonstrated that ΔV varied as S and that Q varied as ΔS so that, from Equation 2 approximately,

$$R = \rho Q(\Delta V) = K_1 (\Delta S) S$$

where k_1 is a constant. Since slip is usually expressed as a proportion of S, this equation becomes

$$R = k_1 S^2(\Delta S/S) \qquad (7)$$

where $(\Delta S/S)$ is the percentage slip.

From Equation 7 it is apparent that the torque varies as the slip, as already demonstrated, and as the square of impeller speed for constant slip percentage. It is apparent that at low input speeds the coupling is relatively "soft." This is why good engagement characteristic at start is obtained with throttle control, such as used on an auto or truck. With tail shaft stalled, full torque will usually be reached at about 40 per cent of full-load speed.

Torque Converters: It was pointed out in the discussion of the fluid coupling that the addition of a third element made possible torque multiplication. Referring to *Fig.* 14, if the turbine vanes continued curving until ΔV over the turbine vanes equaled say, $2S$, the thrust on the turbine would be twice that on the impeller and twice the torque would be developed at the output shaft. However, the discharge V_3 from the turbine would not coincide in direction with V_1. Coincidence could be brought about by adding stator vanes, as in *Fig.* 17 which illustrates a torque converter with stalled turbine. As before, the impeller vanes are mounted on the periphery of a wheel keyed to the input shaft and the turbine similarly keyed to the output shaft. The stator vanes are mounted on a third wheel either anchored to the case in some fashion, or keyed to a third (hollow) shaft, coaxial with the other two and anchored.

The vector diagram in the figure follows the same pattern as before. As shown, the torque multiplication is two but there is no theoretical limitation to this because, by suitable curving of the vanes of all three elements, any value may be obtained. As with the coupling, V_4 at stator discharge is shown greater than V_1; in actual operation it must be equal, the difference representing the fluid losses along the flow path. Due to the longer route and to the fact that considerably more fluid is involved, the rate of flow in the usual design will be much lower at full torque load than with the coupling.

It might be expected that much more energy would have to be wasted to obtain full-load input torque at the output shaft, but, owing to the torque multiplica-

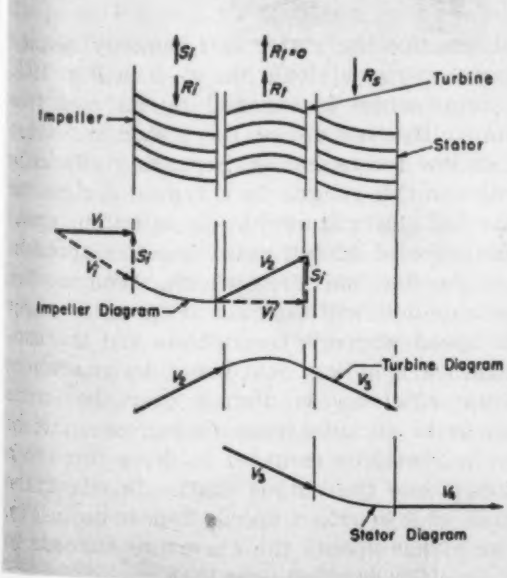


Fig. 17 — Left — Stalled fluid torque converter, showing impeller, turbine and stator diagrams

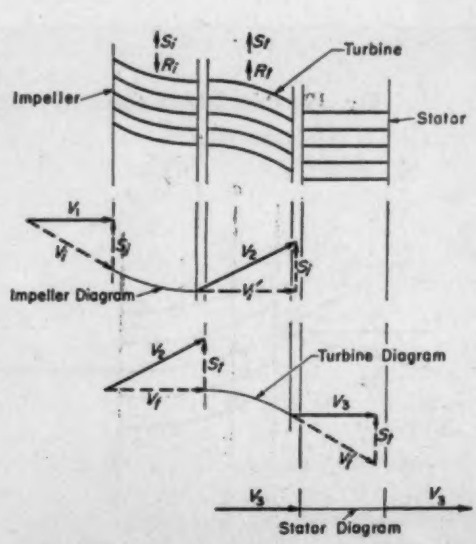


Fig. 18 — Right — Diagram for torque converter when running at impeller speed

tion, this is not so. On a typical coupling full output torque is obtained with input speed of 40 per cent of rated speed. On one typical converter full-load torque is obtained with input shaft speed of 55 per cent for which required input torque is only 25 per cent, so that the energy requirement is only (55/40) .25 = 34 per cent of that for the coupling. Now, this may not be important from an energy (fuel) standpoint, but it may be very important from a waste heat disposal standpoint, in particular if the stalled turbine condition is to persist for any time.

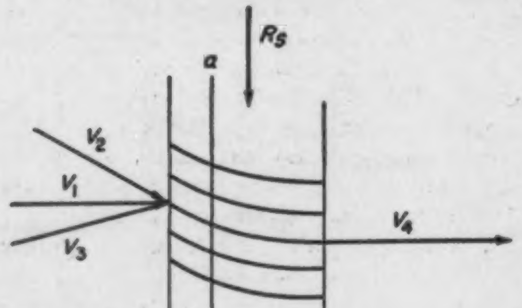
In the foregoing, comparison is made with rated input torque at the output shaft. If comparison were made with a change-speed transmission in connection with the coupling, say a 2 to 1 ratio, against the converter alone, the results would not be so unfavorable. To double the torque on this particular converter, input shaft speed will have to be 72 per cent and input torque 50 per cent, so that waste heat is now (72/40) .50 = 90 per cent of that of the coupling-transmission combination.

Substantially the same relationships will exist in the torque as before. Input torque requirements will vary as the square of input speed. Output torque with turbine stalled will also vary as the square of input speed so that the torque ratio is constant, a conclusion also apparent from consideration of the vector analysis of *Fig.* 17.

The other speed extreme for the converter—the turbine running at impeller speed—is shown in *Fig.* 18. The vector diagram appears the same as for the coupling, *Fig.* 15, in fact is the same except for the stator unit through which the fluid flows without change. Theoretically the converter is now a coupling, and the discussion of the coupling at low slip applies. However, the fluid friction and therefore the slip will be much greater because of the flow through the stator vanes.

In *Fig.* 18 the stator vanes are shown as straight, whereas in an actual unit the stator vanes would, of course, be curved; also the turbine vanes would have more curvature than the impeller, somewhat as in *Fig.* 17. It is then apparent that with an actual converter running as a coupling, vane entrance losses

Fig. 19—Fluid flow through stator



may be very high, flow will be highly turbulent and total fluid loss inherently much more than in a coupling; in fact with early designs coupling range was often reached at a slip as high as 25 per cent, requiring dissipation of a large amount of heat. Efficiency was much better in converter range, the Foettinger unit mentioned in Part 1 reaching 85 per cent at designed operating range, and values of 90 per cent are common today. It should be noted also that the condition becomes worse as stalled torque ratio is increased.

For these reasons it was generally believed at one time that it was impractical to use a converter in the coupling range and that a "lock-up" clutch must be employed if much use was required in this position. That this is still a disputed point is evidenced by the fact that a number of automobile and bus converter designs in current use are so constructed.

Stator Free-Wheels at Coupling Speed

Recent years have seen the introduction of an innovation in stator design that has greatly improved coupling range operation. *Fig.* 19 illustrates fluid flow into the stator only. Under stall conditions, *Fig.* 17, the flow into the vanes is along V_2, and the reaction R_S is developed. At coupling position, *Fig.* 18, the flow is along V_1 and no reaction is developed. At in-between conditions the flow is in a direction between V_1 and V_2 and a reaction in direction R_S, but less in value exists. When the fluid enters at or near the V_1 direction the curved vanes force an abrupt change in flow direction to V_2, with heavy entrance losses. Now, if the stator could be caused to rotate in the direction of the turbine and impeller, the relative flow into the vanes would be inclined more favorably for entrance. This can be permitted by mounting the stator wheel on a one-direction or free-wheeling clutch, so that rotation in the direction R_S is prevented, but allowed in the reverse direction. Then if the direction of discharge from the turbine is such as to afford a small component in reverse direction to R_S, as V_3, for example, the reaction on the stator will be reversed and it will free-wheel along with the turbine wheel, with substantial reduction in entrance losses.

In actual practice the stator is commonly divided into two parts—perhaps along line $a-b$ in *Fig.* 19—and each stator wheel is engaged by its own free wheeling unit. By this device the stator curvature is reduced at low torque ratios, improving efficiency characteristics in this range. In a typical design, for example, the full stator is used up to an output speed of about 65 per cent of full rated impeller speed at which point the first unit free wheels. The smaller curvature stator unit will then act alone up to about 90 per cent speed where it free wheels and the converter becomes a coupling. On some designs where the maximum efficiency is desired over the entire range, such as in an auto transmission, one part of the turbine may also be mounted to drive through a free-wheel unit into the output shaft. In effect, the turbine vanes at low output speeds appear as in *Fig.* 17, while at higher speeds the curvature changes to

(*Concluded on Page* 189)

Calculating Deflection in Circular Plates

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Fig. 1—Magnetic chuck is an example of a thin circular plate loaded by unsymmetrically distributed, perpendicular shearing forces

DETERMINING the stiffness and strength of a thin, circular plate loaded by an unsymmetrical distribution of shearing forces applied perpendicularly to the plane of the plate is a problem of frequent interest to designers. One example occurs in the design of magnetic chucks for rotary grinders. The magnetic chuck, *Fig.* 1, may be regarded as a circular plate mounted on a central hub, and loaded at the edge by pressure of the grinding wheel. Here, the outer edge is not supported. Since the chuck usually rotates rather slowly, stresses in the chuck due to inertia forces would be small compared to those caused by the pressure of the grinder. The problem is to calculate the deflections and stresses in the circular plate because of the unsymmetrical edge loading. The analysis which follows will indicate the procedure to be followed in any problem involving an unsymmetrical distribution of shearing forces or bending moments applied to the edge of a circular plate.

The plate and its loading may be idealized as shown in *Fig.* 2. Applying the usual thin plate theory, it is assumed that (1) the plate is made of a homogeneous, isotropic material that follows Hooke's Law, (2) the deflections of the plate are small compared to its thickness, and (3) the thickness of the plate is small compared to its radius. Because of the circular contour of the plate, it is logical to employ the polar coordinate system, *Fig.* 2. In this system any point in the plate is located by the co-ordinates r and θ. Vertical displacement of any such point, due to the load acting on the plate, is designated by w and is assumed to be positive in the downward direction.

Loading on the plate consists of an arbitrary distribution of shearing forces acting around the outer edge of the plate and applied perpendicularly to the plane of the plate, with V designating the magnitude of this force per unit arc length of the edge of the plate. Because all these forces are at the same distance from the center of the plate, it is apparent that

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the variation of V will depend only upon the angular co-ordinate θ. In order that all possible variations of shearing force may be considered, this applied force may be represented by a Fourier series[1] of the form

$$[V]_{r=a} = -\frac{Et^3}{12(1-\nu^2)}\left[\frac{e_0}{2} + \sum_{n=1}^{\infty}(e_n \cos n\theta + f_n \sin n\theta\right] \quad (1)$$

in which t is the thickness of the plate and ν represents Poisson's ratio. In any problem the applied loading V will be known as a function of θ. This relationship may be written in the form $V = V(\theta)$. The coefficients e_n and f_n in Equation 1 may then be determined by the relations[1]

$$e_n = \frac{1}{\pi}\int_{-\pi}^{\pi} V \cos n\theta d\theta \quad (n = 0, 1, 2, \ldots) \quad (2)$$

$$f_n = \frac{1}{\pi}\int_{-\pi}^{\pi} V \sin n\theta d\theta \quad (n = 1, 2, \ldots) \quad (3)$$

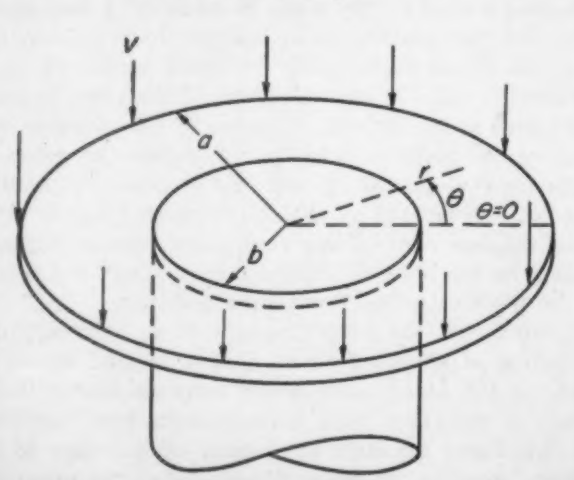
For most engineering applications, a relatively few terms in the series given by Equation 1 will satisfactorily represent the known variation of applied forces.

Timoshenko[2] has demonstrated that the deflection w of a circular plate loaded only at the edge is

$$\begin{aligned} w = {} & A_0 + B_0 r^2 + C_0 \log_e r + D_0 r^2 \log_e r \\ & + (A_1 r + B_1 r^3 + C_1 r^{-1} + D_1 r \log_e r)\cos\theta \\ & + \sum_{n=2}^{\infty}(A_n r^n + B_n r^{-n} + C_n r^{n+2} + D_n r^{-n+2})\cos n\theta \\ & + (A_1' r + B_1' r^3 + C_1' r^{-1} + D_1' r \log_e r)\sin\theta \\ & + \sum_{n=2}^{\infty}(A_n' r^n + B_n' r^{-n} + C_n' r^{n+2} \\ & + D_n' r^{-n+2})\sin n\theta \quad (4) \end{aligned}$$

[1] References are tabulated at the end of the article.

Fig. 2—General problem, of which magnetic chuck is typical, employs polar co-ordinate system in analysis



where the constants A_n, B_n, C_n, D_n, A_n', B_n', C_n' and D_n' are to be determined from the conditions of support and loading.

The general case, such as a magnetic chuck, may be considered to be a flat circular plate with a concentric central hole. The outer and inner radii of the plate are denoted by a and b respectively. Because of the nature of the hub, the plate may be considered to be clamped along the inner edge, $r = b$.

Consideration of these conditions of loading and support leads to the systems of Equations 5, 6 and 7 solved in order to determine the constants A_n, B_n, C_n and D_n in Equation 4:

For case where $n = 0$

$$A_0 + B_0 b^2 + C_0 \log_e b + D_0 b^2 \log_e b = 0 \quad (5a)$$

$$2B_0 b^2 + C_0 + D_0(b^2 + 2b^2 \log_e b) = 0 \quad (5b)$$

$$B_0(2 + 2\nu)a^2 - C_0(1-\nu) + D_0 a^2(3 + \nu + 2\log_e a + 2\nu \log_e a) = 0 \quad (5c)$$

$$4D_0 = \frac{e_0}{2a} \quad (5d)$$

For case where $n = 1$

$$A_1 b^2 + B_1 b^4 + C_1 + D_1 b^2 \log_e b = 0 \quad (6a)$$

$$A_1 b^2 + 3B_1 b^4 - C_1 + D_1 b^2(1 + \log_e b) = 0 \quad (6b)$$

$$B_1(6a^4 + 2\nu a^4) + C_1(2 - 2\nu) + D_1 a^2(1+\nu) = 0 \quad (6c)$$

$$B_1(6a^4 + 2\nu a^4) + C_1(2 - 2\nu) - D_1 a^2(3-\nu) = e_1 a^4 \quad (6d)$$

For case where $n = 2, 3, \ldots$

$$A_n b^n + B_n b^{-n} + C_n b^{n+2} + D_n b^{-n+2} = 0 \quad (7a)$$

$$A_n(nb^{n-1}) - B_n(nb^{-n-1}) + C_n(n+2)b^{n+1} + D_n(2-n)b^{-n+1} = 0 \quad (7b)$$

$$\begin{aligned} & A_n[(n)(n-1)(1-\nu)]a^{n-2} \\ & + B_n[(n)(n+1)(1-\nu)]a^{-n-2} \\ & + C_n[(n+1)(n+2) - \nu(n+1)(n-2)]a^n \\ & + D_n[(n-2)(n-1) - \nu(n+2)(n-1)]a^{-n} = 0 \quad (7c) \end{aligned}$$

$$\begin{aligned} & A_n[(n^2 - n^3)(1-\nu)]a^{n-3} \\ & + B_n[(n^2 + n^3)(1-\nu)]a^{-n-3} \\ & + C_n[(-n^3 + 3n^2 + 4n) + \nu(n^3 + n^2)]a^{n-1} \\ & + D_n[(n^3 + 3n^2 - 4n) - \nu(n^3 - n^2)]a^{-n-1} = e_n \quad (7d) \end{aligned}$$

Two additional sets of equations involving the primed coefficients A_n', B_n', . . . appearing in Equation 4 may be written by analogy with Equations 6 and 7. To form these two sets of equations, A_n is replaced by A_n', B_n by B_n', C_n by C_n', D_n by D_n' and e_n by f_n for all values of n.

To summarize, the procedure for determining the deflection w of a circular plate loaded by an arbitrary distribution of shearing forces around the outer edges is:

1. Express the variation of shearing forces as a function $V(\theta)$
2. Using Equations 2 and 3, calculate the Fourier coefficients e_n and f_n appearing in Equation 1
3. Substitute numerical values of the radii a and b, Poisson's ratio ν, and the coefficients e_n and f_n in Equations 5, 6 and 7 and the analogous equations containing A'_n, B_n, . . .
4. Solve these successive groups of four simultane-

Nomenclature

a = Outer radius of plate, inches
b = Inner radius of plate, inches
Flexural rigidity of plate
D = Young's modulus, psi
M_r, M_θ = Bending moments per unit length of sections perpendicular to radius and tangent, lb inch per inch
$M_{r\theta}$ = Twisting moment per unit length, lb inch per inch
r, θ = Polar coordinate directions
S_r, S_θ = Bending unit stresses acting on sections perpendicular to radius and tangent, psi
t = Thickness of plate, inches
V = Resultant shearing force per unit length, lb per inch
w = Deflection of plate, inches
y = Distance of a point in the plate from the neutral surface of the plate, inches
ν = Poisson's ratio

ous equations, each of which will contain four unknowns. (Equations 5, 6 and 7 could be solved here to yield explicit values of A_n, B_n, . . . but the resulting expressions are rather cumbersome and it is, perhaps, preferable to carry out the solution of each of the groups of equations by immediately substituting numerical data.)

In this manner the deflection w, as given by Equation 4, may be determined to any desired accuracy.

Once the deflection has been determined (Equation 4), the bending moments M_r and M_θ acting on sections perpendicular to a radius and a tangent, respectively, and the twisting moment $M_{r\theta}$ (all per unit length) may be determined from the expressions[2]

$$M_r = -D\left[\frac{\partial^2 w}{\partial r^2} + \nu\left(\frac{1}{r}\frac{\partial w}{\partial r} + \frac{1}{r^2}\frac{\partial^2 w}{\partial \theta^2}\right)\right] \quad (8)$$

$$M_\theta = -D\left[\frac{1}{r}\frac{\partial w}{\partial r} + \frac{1}{r^2}\frac{\partial^2 w}{\partial \theta^2} + \nu\frac{\partial^2 w}{\partial r^2}\right] \quad (9)$$

$$M_{r\theta} = (1-\nu)D\left[\frac{1}{r}\frac{\partial^2 w}{\partial r \partial \theta} - \frac{1}{r^2}\frac{\partial w}{\partial \theta}\right] \quad (10)$$

In these equations,

$$D = \frac{Et^3}{12(1-\nu^2)}$$

Knowing M_r and M_θ, the radial and tangential components of bending stress at any point are given by the equations

$$S_r = \frac{12M_r}{t^3}y \quad (11)$$

$$S_\theta = \frac{12M_\theta}{t^3}y \quad (12)$$

where y is the distance of the point from the neutral surface of the plate.

It should be noted that the satisfactory calculation of the bending moments requires a more accurate expression for w, i.e., more terms in Equation 4, than would be required for a determination of the deflection to the same degree of accuracy. This is because the expressions for bending and twisting moments, Equations 8, 9, and 10, involve the second derivatives of the expression for deflection. This is true of series solutions of most problems involving the bending of plates; convergence of the series representing the moments is always slower than that of the series representing deflections.

Several special cases of this problem have been discussed by Timoshenko[2]: where there is a uniform distribution of shearing forces acting around the outer edge of the plate, and where a single concentrated force is applied at the edge of a plate. The general theory presented in this article readily reduces to the known solutions in both of these special cases.

NUMERICAL EXAMPLE: A circular plate of four-inch outer diameter mounted on a two-inch shaft will be considered. Assume the loading consists of a shearing load uniformly distributed over half of the outer circumference and let V_0 be the intensity of this shear per unit arc length. Such a loading is shown in *Fig.* 3. The loading may be expressed by

$$V = V_0 \text{ for } -\frac{\pi}{2} < \theta < \frac{\pi}{2}$$

$$V = 0 \text{ for } \frac{\pi}{2} < \theta < \frac{3\pi}{2}$$

From Equations 2 and 3, the Fourier coefficients of V_0 may now be calculated:

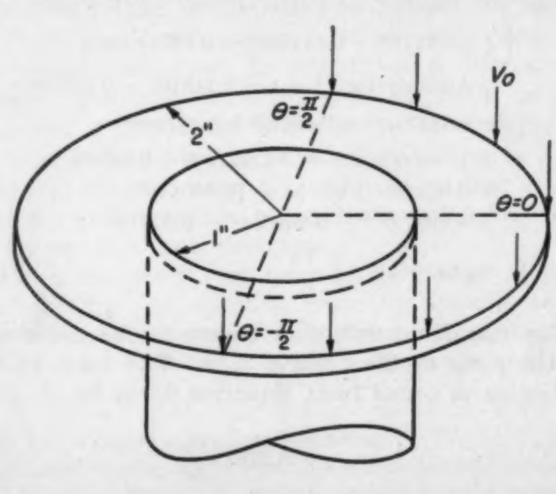
$$e_0 = \frac{2}{\pi}\int_0^{\frac{\pi}{2}}\left(-\frac{V_0}{D}\right)d\theta = -\frac{V_0}{D}$$

$$e_n = \frac{2}{\pi}\int_0^{\frac{\pi}{2}}\left(-\frac{V_0}{D}\right)\cos n\theta d\theta = -\frac{2V_0}{n\pi D}\sin\frac{n\pi}{2}$$

$$f_n = 0$$

Now substitute $a = 2$, $b = 1$, $\nu = 0.3$ and the foregoing values of e_n and f_n in Equations 5, 6 and 7, and

Fig. 3—Example of thin circular plate is loaded uniformly around half of the circumference



in the analogous equations containing A_n', B_n', C_n', and D_n'. The following groups of equations are obtained by taking values of n through 5:

$$A_0 + B_0 = 0 \quad \text{(13a)}$$

$$2B_0 + C_0 + D_0 = 0 \quad \text{(13b)}$$

$$10.4B_0 - 0.7C_0 + 20.40876D_0 = 0 \quad \text{(13c)}$$

$$D_0 = -\frac{V_0}{16D} \quad \text{(13d)}$$

$$A_1 + B_1 + C_1 = 0 \quad \text{(14a)}$$

$$A_1 + 3B_1 - C_1 + D_1 = 0 \quad \text{(14b)}$$

$$105.6B_1 + 1.4C_1 + 5.2D_1 = 0 \quad \text{(14c)}$$

$$105.6B_1 + 1.4C_1 - 10.8D_1 = -\frac{32}{\pi}\frac{V_0}{D} \quad \text{(14d)}$$

$$A_3 + B_3 + C_3 + D_3 = 0 \quad \text{(15a)}$$

$$3A_3 - 3B_3 + 5C_3 - D_3 = 0 \quad \text{(15b)}$$

$$268.8A_3 + 8.4B_3 + 4812.8C_3 - 4D_3 = 0 \quad \text{(15c)}$$

$$-806.4A_3 + 25.2B_3 + 5836.8C_3 + 146.4D_3 = \frac{128}{3\pi}\frac{V_0}{D} \quad \text{(15d)}$$

$$A_5 + B_5 + C_5 + D_5 = 0 \quad \text{(16a)}$$

$$5A_5 - 5B_5 + 7C_5 - 3D_5 = 0 \quad \text{(16b)}$$

$$1433.6A_5 + 21B_5 + 149{,}913.6C_5 + 14.4D_5 = 0 \quad \text{(16c)}$$

$$-71680A_5 + 105B_5 + 61{,}440C_5 + 600D_5 = -\frac{102.4}{\pi}\frac{V_0}{D} \quad \text{(16d)}$$

Three groups of equations analogous to 14, 15, and 16 with the A_n, B_n, C_n, and D_n replaced by the corresponding primed quantities may also be written. The right side of each of these equations is zero, hence the primed coefficients are each equal to zero for this loading.

The groups of Equations 13, 14, 15, and 16 are solved to yield numerical values of the coefficients A_n, B_n, C_n, and D_n. Substituting these in Equation 4 to give the equation of the deflection surface of the plate,

$$w = \Big\{ [-0.11180 + 0.11180r^2 - 0.16111 \log_e r - 0.06250r^2 \log_e r] + [-0.24810r - 0.03510r^3 + 0.28321r^{-1} + 0.63662r \log_e r] \cos\theta + [-0.00892r^3 - 0.01621r^{-3} + 0.00055r^5 + 0.02458r^{-1}] \cos 3\theta + [0.00044r^4 + 0.00172r^{-5} - 0.00001r^7 - 0.00215r^{-3}] \cos 5\theta + \ldots \Big\} \frac{V_0}{D} \quad \text{(17)}$$

The maximum deflection occurs at the outer edge of the plate at the point $\theta = 0$. The value of this deflection is found from Equation 17 to be

$$w_{\max} = 0.2671 \frac{V_0}{D}$$

This value was obtained by using values of n through 5. Value of the maximum deflection is changed by less than one per cent if the next group of terms, resulting from taking $n = 7$, is considered. Thus, the series converges sufficiently rapidly for practical use.

Fourier Coefficients for Several Loadings: (1) If in *Fig.* 3, the uniform load of intensity V_0 per unit arc length is applied along an arc betwen $\theta = -\alpha$ and $\theta = \alpha$ (instead of between $-\pi/2$ and $\pi/2$), the equation for the loading is

$$V = V_0 \text{ for } -\alpha<\theta<\alpha$$

$$V = 0 \text{ for } \alpha<\theta<(2\pi - \alpha)$$

The Fourier coefficients as given by Equations 2 and 3 become

$$e_o = -\frac{2V_0\alpha}{\pi D}$$

$$e_n = -\frac{2V_0}{n\pi D}\sin n\alpha$$

$$f_n = 0$$

(2) If a load with intensity varying from a maximum value of V_0 at $\theta = 0$ to zero at $\theta = \pm\alpha$ is applied along the arc between $\theta = -\alpha$ and $\theta = \alpha$, the equation for the loading is

$$V = V_0\left[1 - \frac{\theta}{\alpha}\right] \text{ for } -\alpha<\theta<\alpha$$

$$V = 0 \qquad \text{for } \alpha<\theta<(2\pi - \alpha)$$

The Fourier coefficients become

$$e_0 = -\frac{V_0\alpha}{\pi D}$$

$$e_n = -\frac{2V_0}{n\pi D}\sin n\alpha + \frac{2V_0}{n^2\pi D\alpha}[\cos n\alpha + (n\alpha)\sin n\alpha - 1]$$

$$f_n = 0$$

Most practical design problems can be solved by one of these two types of loading.

References

1. R. V. Churchill—*Fourier Series and Boundary Value Problems*, McGraw-Hill Book Co. Inc., New York, 1941, Chapter IV.
2. S. Timoshenko—*Theory of Plates and Shells*, McGraw-Hill Book Co. Inc., New York, 1940, Chapter VII.

They Say...

"The challenge of 1951 is peculiarly directed to science and industry. The technical competence of American scientists and the productive genius of American industry are the indispensable ingredients for success of the political, military, and economic actions already launched, or in prospect. To meet this challenge, science and industry will need to pool their talents and combine their efforts more comprehensively and more incisively than ever before. Full advantage must be taken of all knowledge and 'know-how' already gained. New knowledge and new techniques must be developed with urgent pace. Exchange of experience and of thinking is the essential catalyst to accelerate this process."—Paul E. Pihl, *Rear Admiral, USN.*

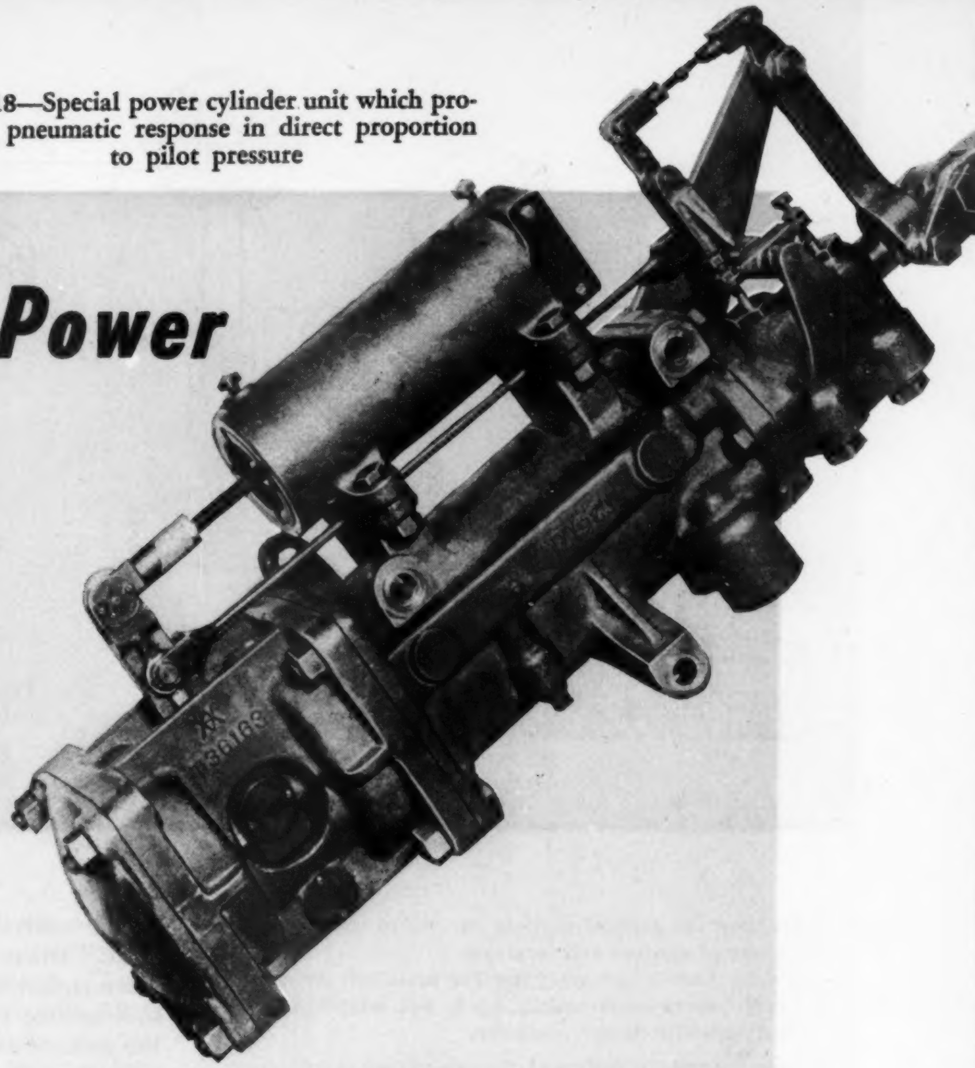


Fig. 18—Special power cylinder unit which provides pneumatic response in direct proportion to pilot pressure

Pneumatic Power

Design and application fundamentals for machine drives and controls

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Part 3—Selection of Components

THERE is a wide variety of common and unusual devices available from which pneumatic systems can be built, *Fig.* 18. Some knowledge of these devices and their characteristics will assist greatly in selecting the best components for specific system requirements. Basically, pneumatic control systems can be divided into three categories composed of devices that can be classified as: (1) Senders, (2) intermediate devices, and (3) receivers. Functionally, the senders initiate action, the receivers produce the result, and the intermediate devices interlock or modify action between the two. In some cases, control may be so simple that intermediate devices are unnecessary.

Each of these divisions will be covered briefly and applications of typical devices of each type will be discussed. The examples are picked from the 100 psi pressure zone as this is most commonly used on machine control systems.

SENDERS: Senders are the operating or controlling devices which control the supply to and from the receivers or actuating devices. The simplest form is the on-and-off type which either delivers full supply pressure to the actuator or exhausts air from it. *Fig.* 19 shows two typical valves of this type. The rotary valve consists of two disks lapped to an airtight fit. One disk rotates on the other to connect the various ports at the lapped surface. This rotary valve construction lends itself to multifunction valves with several positions connecting various ports in each position. The pilot valve is a poppet type supply and exhaust valve, lever-operated. In addition to hand-operated senders, there are many others that are operated by foot pedals, pushbuttons, solenoids, cams, levers and other machine parts.

There are many variations in the design of pneumatic valves and selection generally depends on the particular details involved in the specific system required. Some of the common valve types and their uses are:

1. Ball valves—simple, self-cleaning
2. Disk valves—either metallic or nonmetallic used in wide variety of devices including compressors
3. Wing type valve—both conical and spherical seats

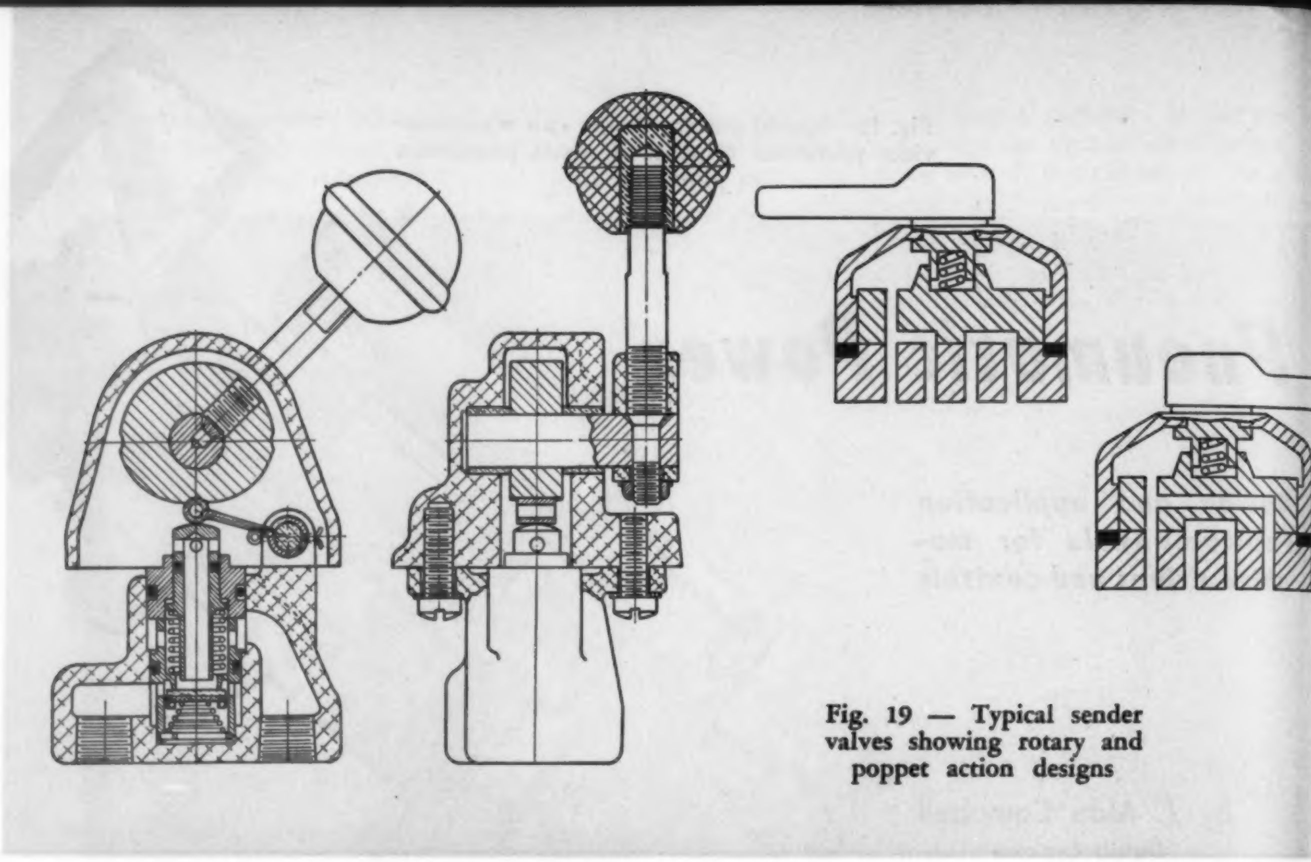


Fig. 19 — Typical sender valves showing rotary and poppet action designs

are used for general-purpose valving in checks and variety of senders and receivers

4. Rotary valves—greatest use for multiport senders
5. Spool valves—less widely used, but adaptable to many specific design problems.

Another entirely different functional principle from the on-and-off valve is offered by the pressure-control valve, *Fig.* 20. The handle position on this valve determines the pressure output. Each handle position corresponds to a particular delivery pressure so that, for instance, the quadrant of handle movement might be calibrated from 0 to 100 psi. For any particular handle position the valve will not only deliver the corresponding pressure for that position but it will deliver additional air or exhaust air from the line so as to maintain that pressure so long as the handle remains in the same location.

The pressure-control valve is used to operate brakes, throttles, governors, or any devices that require a variable pressure. It either controls the output force as when it pilots an air cylinder or it controls position as when it pilots a positioner such as the throttle actuator.

An interesting application of the pressure-control valve in the machine shop is the association of the valve with a pneumatic chuck on a lathe. Here the valve can be set to provide a high gripping force in the chuck during the initial rough cuts on a production item. As the machining progresses, the pressure can be reduced so that it will be low at the final finish operations to minimize distortion. The same principle is used for other machine chucks and vises.

Various combinations of pressure-control valves and on-and-off valves can be built into a single housing for special purposes. In *Fig.* 21 is shown a valve containing two elements, one pressure control type and one on-and-off. This might be used for single handle control of a clutch and brake on a crane or hoist. Other variations include from one to four valve elements, any number of which can be the pressure-control type. This multiple type valve reduces the number of control handles and lends itself to interlocking functions so they cannot be performed simultaneously.

RECEIVERS: Receivers or actuating devices include single and double acting cylinders, *Fig.* 22, which are used for many purposes. They vary in diameter from less than an inch to several feet, and have strokes of almost any reasonable value. Cylinders with strokes of 15 feet or more are not uncommon. Where very short strokes are required and an absolute minimum of friction is desired, a rubber or metal diaphragm can be used.

Accurate Positioning Possible

Where accurate positioning is required there are several types of positioning actuators. The simplest is a cylinder or diaphragm backed by a heavy spring. Where greater forces are required a positioner such as the Pneudyne shown in *Fig.* 18 can be used. This consists of a power cylinder which will assume any position in direct proportion to the amount of pilot pressure. In principle the pilot pressure is applied to a diaphragm or piston which unbalances the supply and exhaust valves. These valves flow air to or from the power piston until it moves to the position corresponding to pilot pressure. At this point a mechanical linkage from the power piston returns the valves to balance. It is characteristic of such positioners that they will, within their rating, maintain the position indicated by pilot pressure independent of variations of the external force or forces against which they operate.

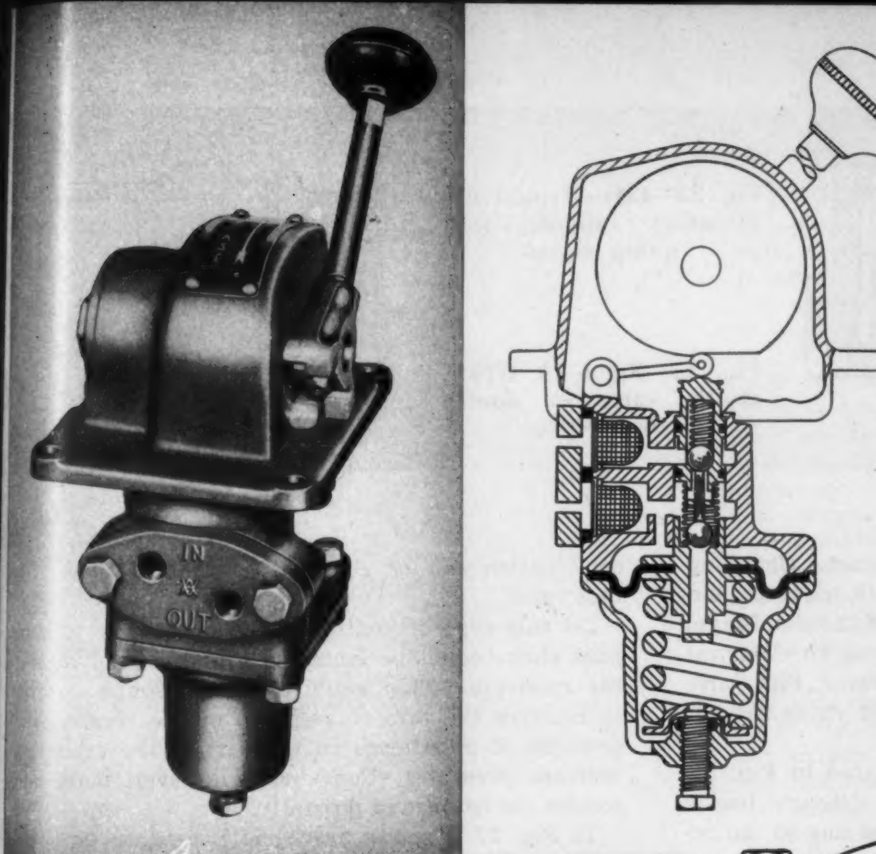


Fig. 20—Left—External and diagrammatic view of a pressure-control valve. Pressure varies according to the handle setting

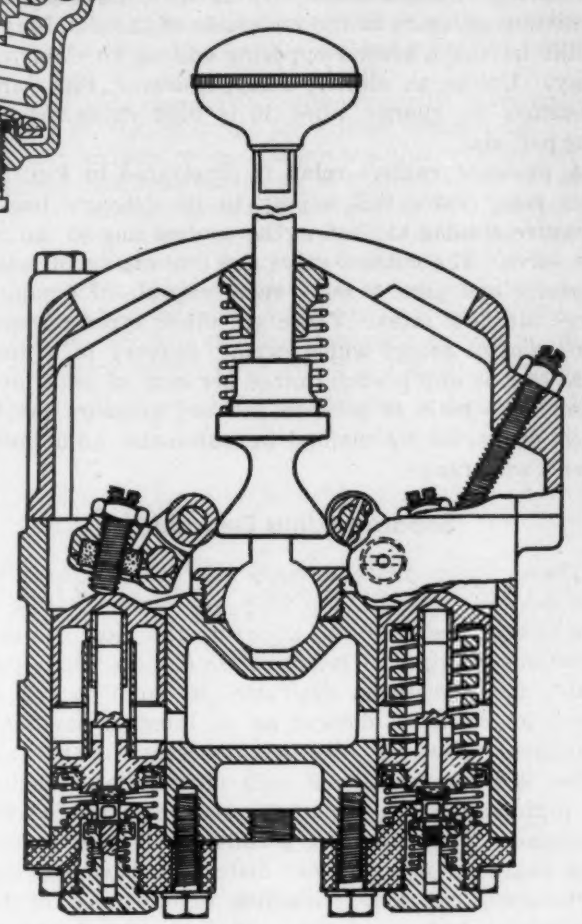


Fig. 21—Below—Special-purpose combination valve containing a pressure-control and an on-and-off unit for dual functions

Another positioning arrangement is to use a rotary air motor with valves that cut off supply pressure just as limit switches are used on electric motors. This is useful where rotary motion is involved and where angular rotation may be beyond the limits of linear relationship either directly or through gear sectors.

INTERMEDIATE DEVICES: Numerous intermediate devices are available to perform special functions in pneumatic systems. Five of these will be described to illustrate typical functions.

The check valve is the simplest intermediate device. It is used to prevent backflow through a line. Frequently it may have a by-pass orifice so that full flow is permitted in one direction and restricted flow in the opposite direction.

The shuttle valve, also known as a double check valve, is illustrated in *Fig.* 23. This particular valve permits pressure from either of two pilot lines to be delivered to a cylinder or other device. Pressure takes precedence over exhaust, which means that the pilot line having the higher pressure would supply the cylinder and move the shuttle to close off the other pilot line. Another type of shuttle valve is one in which exhaust takes precedence over pressure, and the pilot line of lower pressure controls the cylinder.

The quick release valve is a device for obtaining a rapid exhaust of pressure even though the pilot line may be long or small. It is located at the receiver end of the pilot line. When air is supplied through the pilot line it acts as a nonreturn check valve opening to permit flow to the receiver. When air is exhausted from the pilot line the check closes and at the same time opens an exhaust port in the quick release valve. This permits the air from the receiver to exhaust locally rather than have to return through the long or small pilot line.

One of the most versatile intermediate valves is illustrated in *Fig.* 24. This relay valve consists of a diaphragm and return spring, connected to a double valve arrangement. Pilot pressure on the diaphragm compresses the spring and operates supply and exhaust valves so as to open one and close the other. Exhaust of pilot pressure permits the spring to return the diaphragm and reverse the setting of the supply and exhaust valves.

The function of this valve can best be explained by comparing it to an electrical relay. It can be connected so that presence of pilot air permits supply air to flow through the valve to the device it controls. This is like an electrical relay with a front contact. Reversing the supply and exhaust connec-

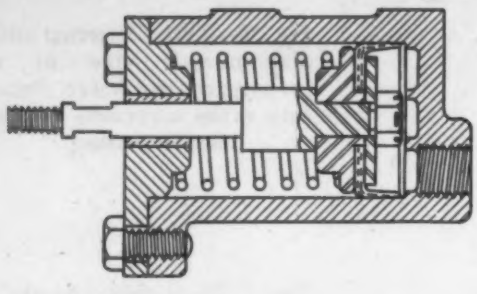


Fig. 22—Left—Typical single-acting cylinder with spring return

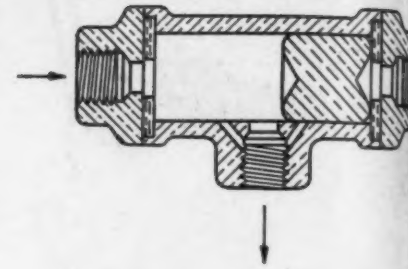


Fig. 23—Right—A typical shuttle valve or double check valve

tion makes it a relay with a back contact. Changing the spring changes sensitivity as with a relay; and admitting pressure to the underside of the diaphragm is like having a second opposing coil on an electrical relay. Unlike an electric relay, however, this valve consumes no energy when it is held energized for long periods.

A pressure control relay is illustrated in *Fig.* 25. This relay valve will supply to its delivery line a pressure similar to that in the control line to the relay valve. The control valve and line can be of small capacity and pilot a relay valve capable of handling large air flow rates. There are other types of pressure control relays which permit delivery of a pressure that is any predetermined per cent of pilot pressure. The ratio of pilot to delivery pressure can be fixed or varied by manual or automatic adjustment over a wide range.

Separate Units Desirable

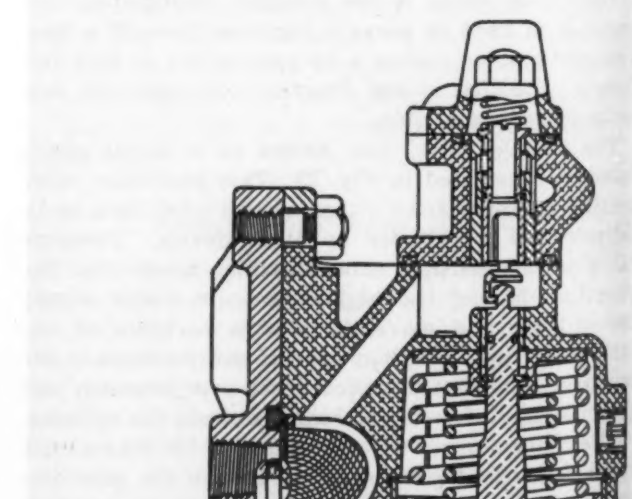
These intermediate devices are by no means all the devices available, but they are representative of the basic types used and illustrate the most common functions required of intermediate devices. It is possible, and sometimes desirable, to build in one or more intermediate devices as an integral part of a machine. However, the overall design of the machine is more versatile if each intermediate function is performed by a separate valve mounted on the machine. This not only permits variations in control characteristics without disturbing basic machine structures, but also simplifies maintenance of the pneumatic devices as they can be removed and replaced as separate units.

INTERLOCKING: Intermediate devices can be used for interlocking. *Fig.* 26 shows a system having positioning provided by remote control. The manually operated pressure control valve, with spring release, for accomplishing this objective is shown at the top of the diagram. Just below it is a controller of the same type, which can be operated by hand or foot. The control lines from these two valves are connected to opposite ends of a shuttle valve, so that either control can be used at will to graduate the receiver. The third arrangement in *Fig.* 26 suggests hand controls for three positioning receivers with intermediate devices added in the form of relay valves, *Fig.* 24, so that if certain lines are energized to bring about a master co-ordination between the receivers, the total combination can be controlled by either a hand or foot valve.

On this type of control, a check valve with a bypass choke could be installed in the control lines to the receivers. This would provide a simple solution to limiting the rate of response of the receivers regardless of impatience on the part of the senders to increase pressure. Conversely, however, decreasing results can be secured promptly.

In *Fig.* 27 is shown a schematic diagram of a control system involving two operations that are to take place in sequence. Example of the two operations might be: (1) Clamping and bending, (2) clamping and cut-off and (3) any other comparable common functions. On the diagram, the two operations are identified as "1st operation" and "2nd operation".

Fig. 24—Pilot-operated relay valve



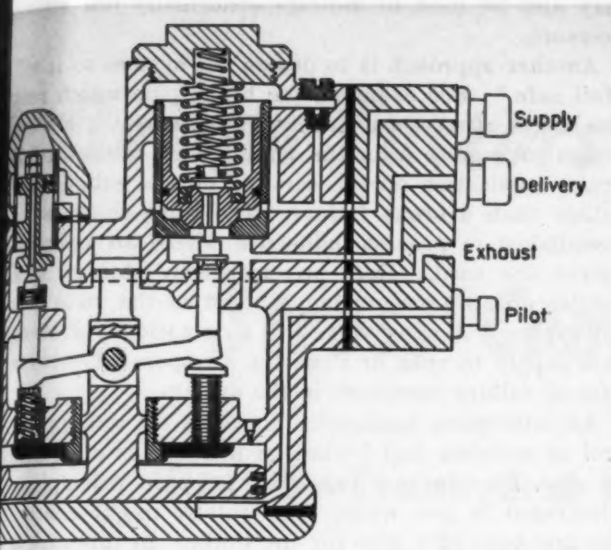


Fig. 25—Presure-control relay valve

Some of the unusual characteristics of the system are: (1) It is a one-cycle system, meaning that only one cycle can be obtained, regardles of how long the starting pushbutton is depressed; (2) the initiation of the cycle sets up a "lock-up" function which is not released until the cycle is complete; and (3) it illustrates the use of an air spring principle as a substitute for release springs in cylinders.

Physically the component parts of the system are:

1. The starting button is a valve similar to the pilot valve shown in *Fig.* 19, except that it has a pushbutton instead of a lever
2. The single-shot feature is provided by relay valve as illustrated in *Fig.* 25
3. The lock-up feature consists of a double check valve, *Fig.* 24, above another relay valve. The latter is of lower operating pressure value than that used for the single-shot feature. It serves to initiate the first operation and further to provide a circuit through a normally-open valve at the right and on back to the other end of the double check valve, previously identified, to establish a lock-up or stick-up function on the initiating relay
4. The second step relay is another relay valve used to initiate the second operation, after the pressure in the first operation has reached a predetermined value
5. The cycle completion feature is a pilot valve similar to *Fig.* 19 except for a lever arrangement
6. The uniform release feature is a true pressure control type reducing valve, in the sense that it can vent exhaust pressures as well as hold pressure up to a desired value. It provides a constant low return force on the cylinder pistons, in lieu of a changing force common to a spring as it is compressed. One valve will serve several cylinders. Physically it is like the regulating portion of the pressure control valve shown in *Fig.* 20.

In operation, when the pushbutton is depressed, the

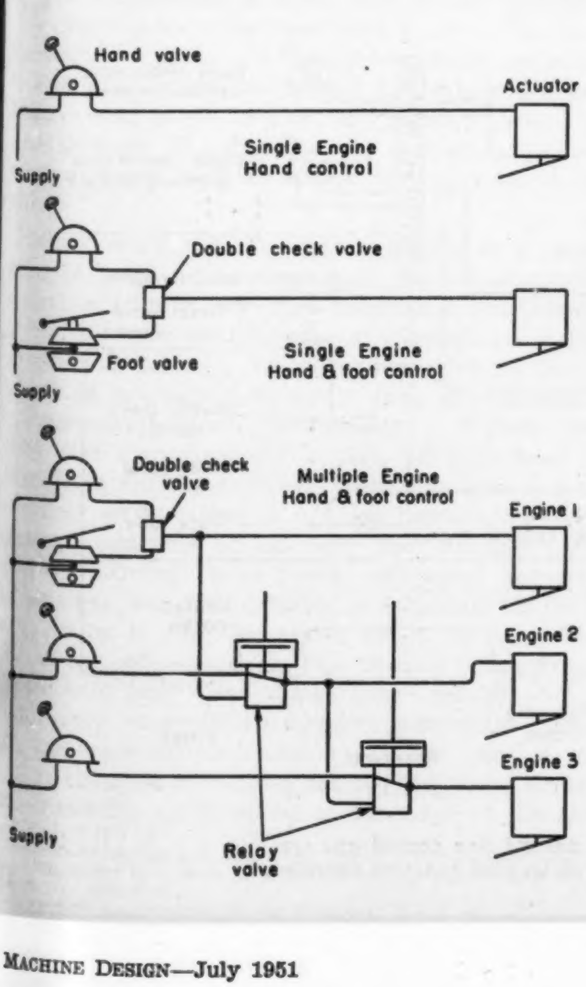
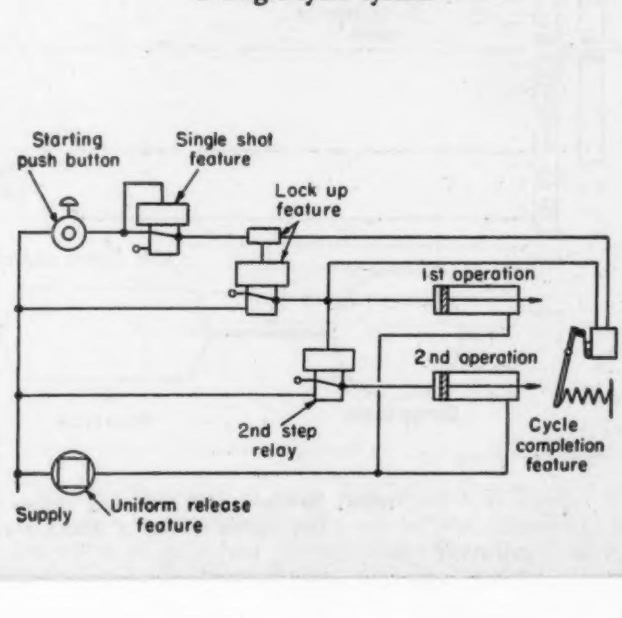


Fig. 26—Left—Diagram of remote throttle control for an oil rig. Multiple engines are coordinated so that any one of the controls may be used

Fig. 27—Below—Schematic circuit diagram for a single-cycle system



single-shot feature will initiate the first operation but will cut off communication to prevent a second cycle until the button is released and pushed again. The lock-up feature performs the double duty of receiving the command for the first operation and then remaining applied until the full cycle is completed. Other illustrations of interlocking might be included, but the field is so vast that space in this article does not permit further expansion.

Emergency Protection: As with any control medium, the designer in pneumatics should always examine what would happen if the air supply failed. With a properly designed supply system there should be no fear of failure. However, some control systems are so vital that additional precautions should be taken.

First the storage reservoirs can be large enough to continue normal operation for sometime after the compressor is stopped. If a single supply system is used for many separate operations throughout a plant, any particularly vital operation can be protected by a local storage tank with a check valve at its inlet connection. This will prevent a failure elsewhere in the system from suddenly depleting the air supply at this point.

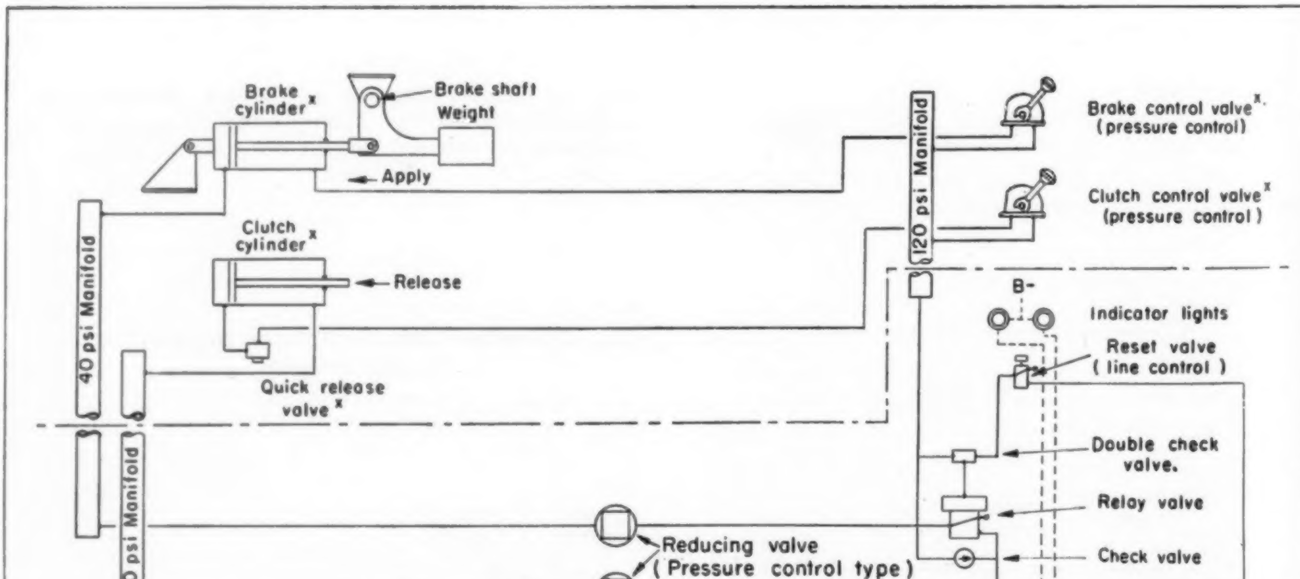
Where practical a stand-by compressor can be used to take care of abnormal air requirements or shutdown of the main compressor for any cause. A pressure switch that operates a warning light or buzzer may also be used to indicate abnormally low supply pressure.

Another approach is to design the system so it will "fail safe." For example, the brake pipe which runs the length of a railway train will be separated if a car comes uncoupled from the train. This causes a full brake application on the car and on the entire train rather than a brake failure. The same thing is accomplished in process industries where air operated valves are used. Here an emergency storage tank bottles up air pressure independent of the main supply system. A relay valve will automatically draw on this supply to open or close the air-operated valve in case of failure elsewhere in the system.

An interesting application concerns the remote control of clutches and brakes on a hoist, as suggested by *Fig.* 28. In this case, the fail-safe principle is illustrated in two ways. One failure might concern the breakage of a pipe for the brake. In this case, a weight would apply the brake. If deficiency in the air supply were to develop, clutches would be released and the brakes would automatically apply. The normal operating conditions could only be secured by a deliberate act of resetting the controls.

Fundamentally, compressed air is a reliable and versatile source of power control. As the designer becomes better acquainted with its characteristics, the field of application of pneumatics continues to expand.

Fig. 28—Schematic of a "fail safe" system for clutches and brakes on a hoist



MACHINE DESIGN
Data Sheet

Design Factors for Stress Concentration

Part 5—Transverse Holes in Bars and Shafts

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CONCLUDING the current series of data sheets on stress-concentration, this article summarizes factors for transverse holes in bars and shafts. Examples of parts with transverse holes are shown in *Fig.* 23. General calculation methods and nomenclature were given in Part 1, February issue. The usual custom of using the net section in defining nominal stress and K factors is continued in this article.

ORIGIN OF CURVES: For tensile loading of a rectangular bar with a transverse hole, *Fig.* 24, stress-concentration factors have been determined photoelastically.[1,2] For the bending case of a round shaft with a transverse hole, *Fig.* 25, stress-concentration factors have been determined by strain gage measurements.[3]

Stress-concentration factors for a finitely wide plate with a transverse hole, *Fig.* 26, have been derived from various sources. Values at $a/w = 0$ for relatively large values of a/h are based upon mathematical solutions of Goodier[4] and Reissner[5] who analyzed infinitely wide plates, with holes, subjected to bending. For finite widths and various values of a/h, curves in *Fig.* 26, drawn by A. M. Wahl and the author, are based on *Fig.* 25 and faired to fit photoelastic test[6,7] and strain gage[8] results.

For pure shear of an infinite plate with a hole, the maximum stress (tension) at the edge of the hole is equal to four times the applied shear stress[9]. The maximum shear stress at the edge of the hole

Curve sheets appearing in this article, together with additional design data, will appear in a forthcoming book to be published by John Wiley & Sons Inc., New York.

[1] References are tabulated at end of article.

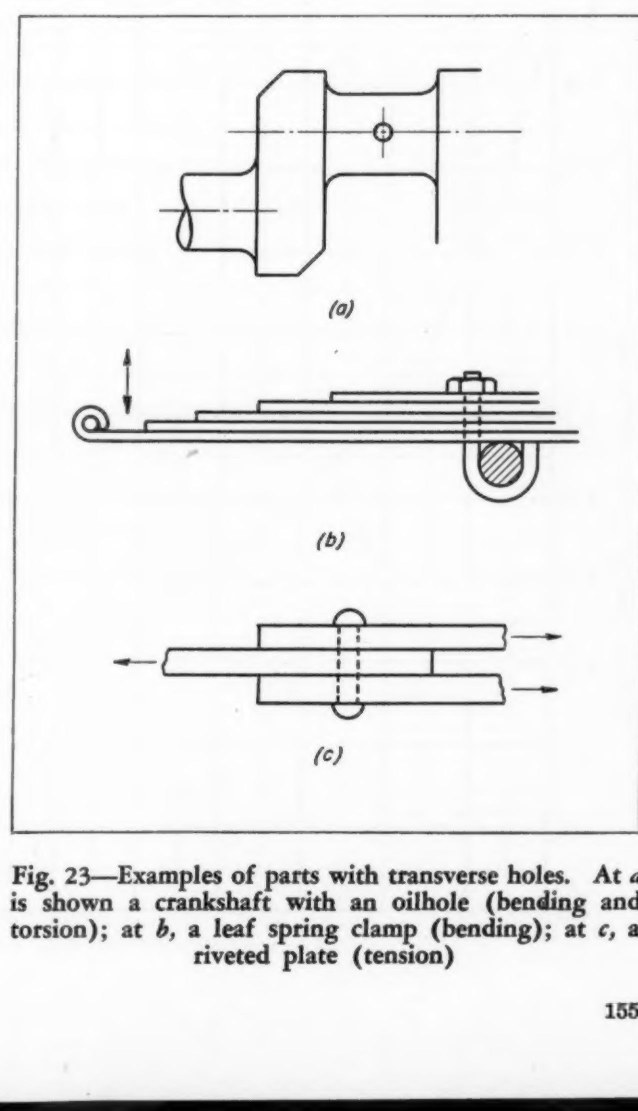


Fig. 23—Examples of parts with transverse holes. At *a* is shown a crankshaft with an oilhole (bending and torsion); at *b*, a leaf spring clamp (bending); at *c*, a riveted plate (tension)

Stress Concentration

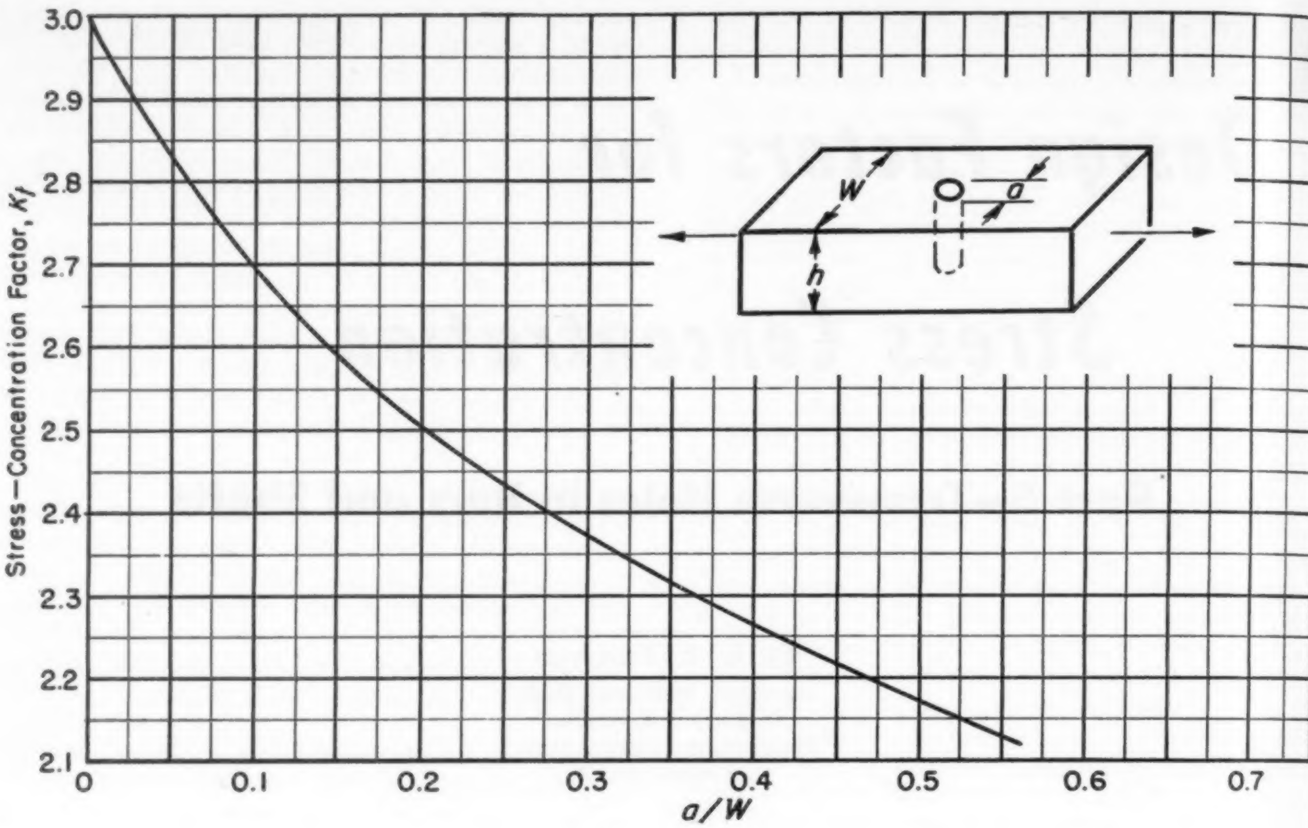
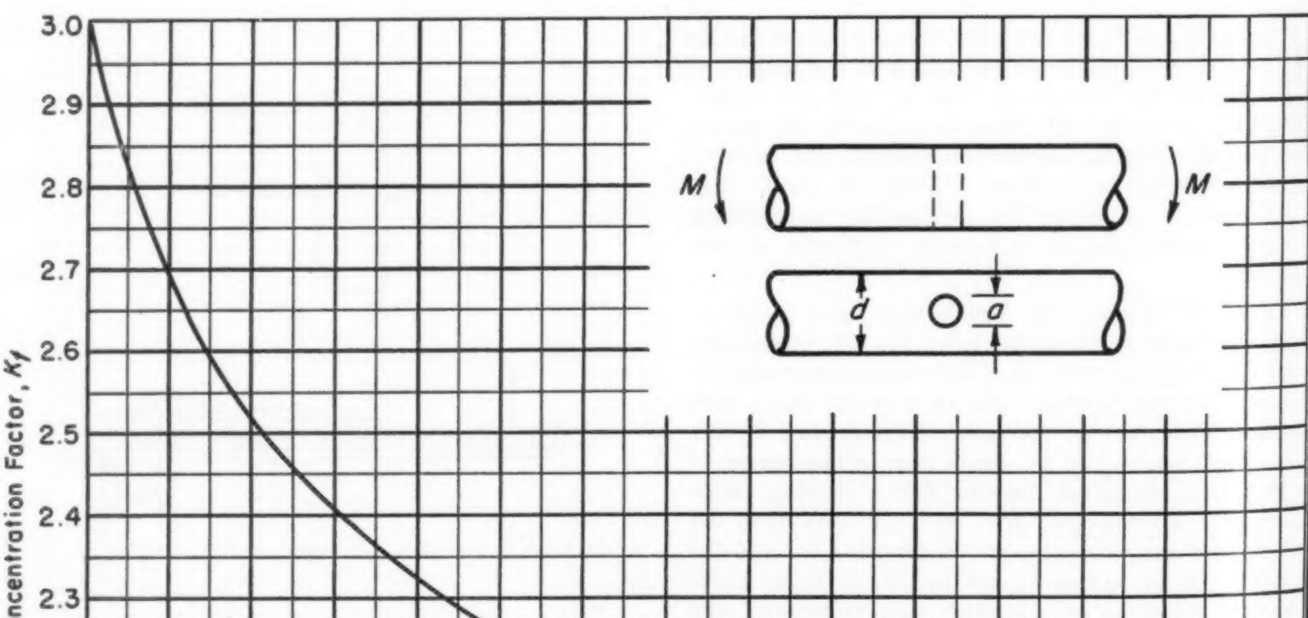


Fig. 24—Above—Stress-concentration factor, K_t, for axial loading of a bar with a transverse hole

Fig. 25—Below—Stress-concentration factor, K_t, for a shaft, with a transverse hole, in bending



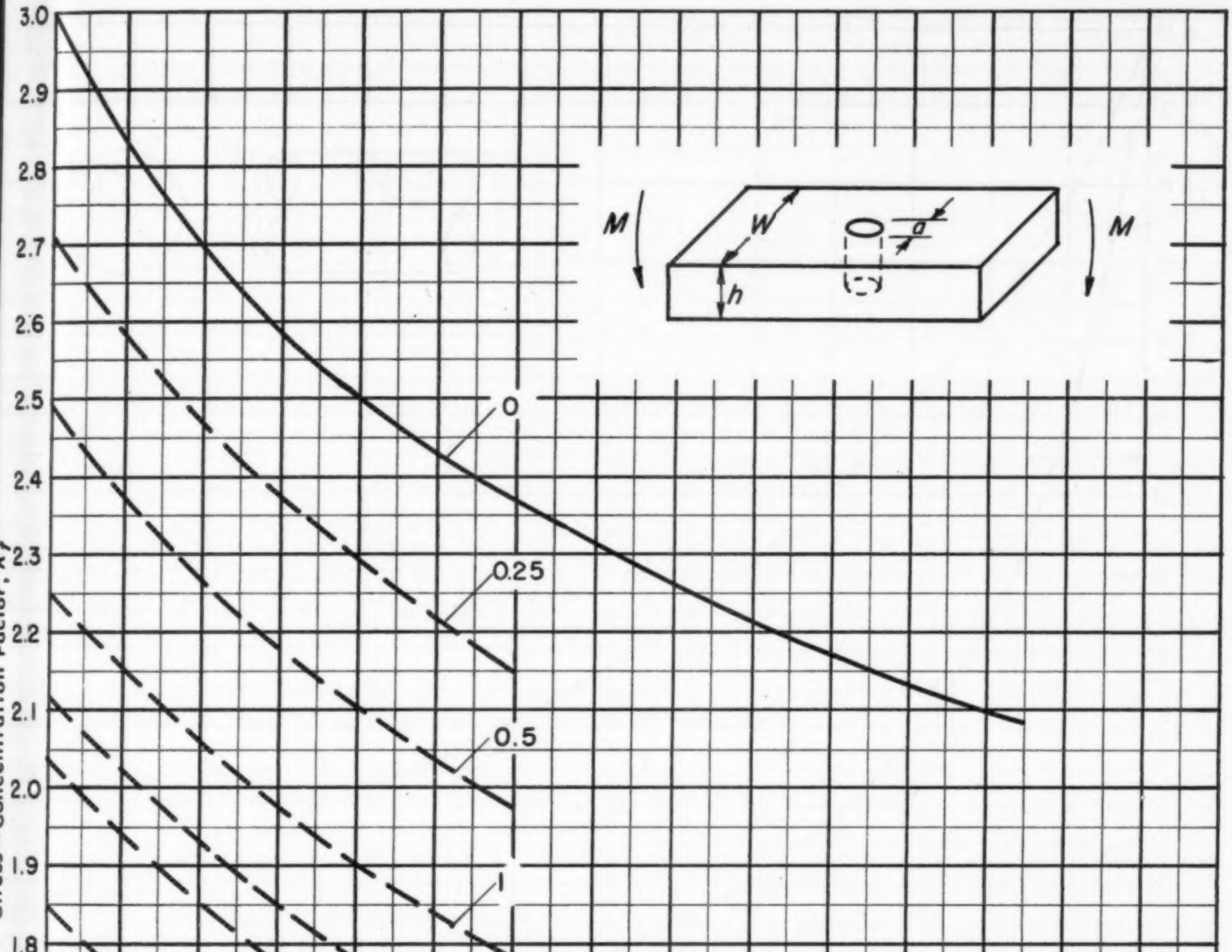


Fig. 26—Stress-concentration factor, K_t, for a flat bar, with a transverse hole in bending

Data Sheet

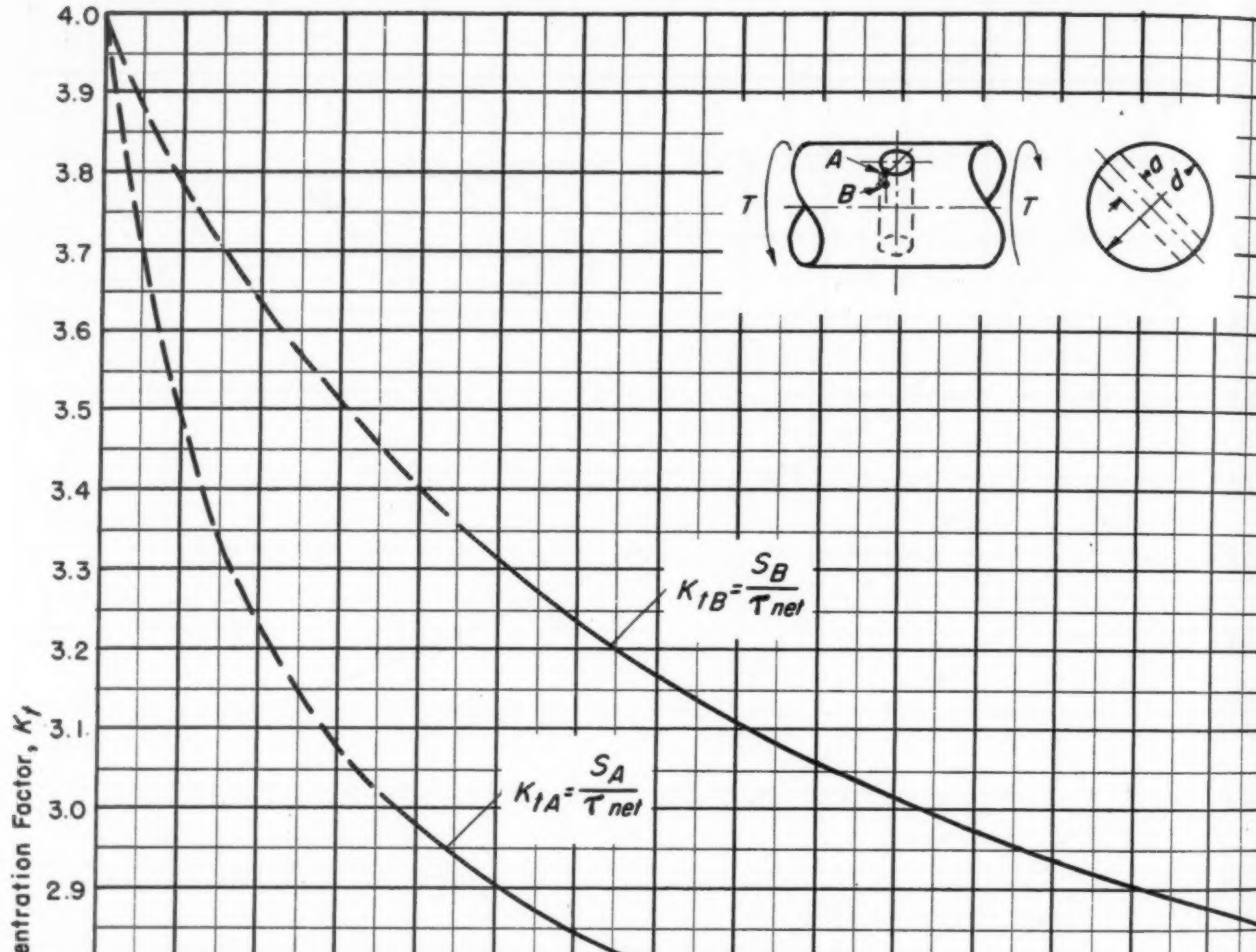


Fig. 27—Stress-concentration factor, K_t, for a shaft, with a transverse hole, in torsion

twice the applied shear stress. For round shafts he values shown in *Fig.* 27 have been obtained by hum and Kirmser[10] with special strain gages. In ig. 27, K_t is arbitrarily defined and the maximum ormal stress (tension) at the hole divided by the pplied shear stress based on net section. That is, $K_{tA} = S_A/\tau_{net}$, $K_{tB} = S_B/\tau_{net}$. Note that the tension higher on the hole surface at a distance slightly elow the shaft surface, at B in *Fig.* 27. Hence, for esign $K_{tB} = S_B/\tau_{net}$ should be used.

These factors should be modified in accordance ith the fatigue-notch factor equation and *Fig.* 5 f Part 1, February issue, with the value of r taken s the radius of the hole, $a/2$. Procedures to be aplied for different types of both loading and maerials are given in the following sections.

TENSION OR BENDING: For brittle material, use K_t for all cases.

For ductile material, use $K_t = 1$ (no stress-conentration effect) for static loading, except as noted n Part 1. For variable loading with ductile material, se K_t values given in this article. Stress-concentraion factor K_t and combined stress-concentration and hear-energy factor K_t' are equal for a transverse ole in tension, or in bending.

TORSION: For brittle material, two theories[11] of ailure are of interest: maximum stress theory and Mohr theory. Certain test data seem to substantiate he former[12] while other tests and reasoning lead to perference for the latter.[13, 14] For the maximumtress theory, K_{tB} of *Fig.* 27, and for the Mohr theory, $(K_{tB}\ S_{uc})/(S_{uc}+S_{ut})$ should be used, where S_{uc} = compressive ultimate strength and S_{ut} = tensile ultimate strength. Note that the former gives a higher actor and, if used in design, will be on the safe side. A premise of the foregoing factors is that Hooke's aw is followed; for curved stress-strain diagrams (cast iron) the factors will be somewhat high, again n the safe side.

For ductile material, the shear-energy theory[15] is mployed. Use $K_t = 1$ for static loading. For variable oading $K_{tB}' = \tau_e/\tau_{nom} = K_{tB}\sqrt{1-C+C^2}/\sqrt{3}$, where K_{tB} is obtained from *Fig.* 27 and C = ratio of principal tresses at B. For the proportions of greatest inerest, C was found[10] to be approximately 0.12. With his value, the above expression reduces to $K_{tB}' =$ 0.545 K_{tB}. For the usual proportions, K_{tB}' is greater han $K_{tA}/\sqrt{3}$. In other words, failure can be expected o start in the hole beneath the shaft surface. This s true for both brittle and ductile materials.

EXAMPLE: Assume a ductile shaft steel (heattreated) with a torsional endurance limit of 29,000 psi. If this is not known, a good assumption is to divided the bending endurance limit by $\sqrt{3}$; that is, $50{,}000/\sqrt{3} = 29{,}000$ psi. If neither endurance limit is known, divide the tensile strength by $2\sqrt{3}$; that is $100{,}000/2\sqrt{3} = 29{,}000$. Let shaft diameter $d = 2$ inches and transverse hole diameter $a = 0.2$-inch $(a/d = 0.1)$. Find the alternating torsional moment carrying capacity of the shaft.

At point B in *Fig.* 27, $\tau_{nom} = \tau_e/K_{tB}' = \tau_e/0.545$ $K_{tB} = 29{,}000/(0.545)(3.31) = 16{,}000$ psi. This is the estimated torsional fatigue strength of the shaft with the transverse hole. From *Fig.* 5, Part 1 (February issue), it can be seen that, for $r = 0.1$-inch, substantially full notch sensitivity can be expected in a heat-treated steel.

Illustrating that point B represents the weakest location, calculations for point A are also shown although normally this calculation need not be made. At point A, $\tau_{nom} = \tau_e/K_{tA}' = \tau_e/0.577\ K_{tA} = 29{,}000/(0.577)(2.90) = 17{,}200$ psi. The approximate net torsional section modulus of the shaft for a reasonably small hole is $J/c = (\pi d^3/16) - (ad^2/6)$. The alternating torsional moment carrying ability of the shaft is $M = \tau J/c = (16{,}000)\ (1.437) = \pm 23{,}000$ inch-pounds.

REFERENCES

1. M. M. Frocht—"Factors of Stress Concentration Photoelastically Determined," *Trans. ASME*, Vol. 57, 1935, Page A67.
2. A. M. Wahl and R. Beeuwkes—"Stress Concentration Produced by Holes and Notches," *Trans. ASME*, Vol. 56, 1934, Page 617.
3. R. E. Peterson and A. M. Wahl—"Two and Three Dimensional Cases of Stress Concentration and Comparison with Fatigue Tests," *Trans. ASME*, Vol. 58, 1936, Page A15.
4. J. N. Goodier—"Influence of Circular and Elliptical Holes on Transverse Flexure of Elastic Plates," *Philosophical Magazine*, Vol. 22, 1936, Page 69.
5. E. Reissner—"The Effect of Transverse Shear Deformation on the Bending of Elastic Plates," *Trans. ASME*, Vol. 67, 1945, Page A69.
6. J. N. Goodier and G. H. Lee—"An Extension of the Photoelastic Method of Stress Measurement to Plates in Transverse Bending," *Trans. ASME*, Vol. 63, 1941, Page A27.
7. D. C. Drucker—"The Photoelastic Analysis of Transverse Bending of Plates in the Standard Transmission Polariscope," *Trans. ASME*, Vol. 64, 1942, Page A161.
8. C. Dumont—NACA Technical Note 740, 1939.
9. S. Timoshenko—*Strength of Materials*, D. Van Nostrand Co., New York, Part II, second edition, 1941, Page 317.
10. A. Thum and W. Kirmser—"Ueberlagerte Wechselbeanspruchungen, ihre Erzeugung und ihr Einfluss auf die Dauerhaltbarkeit und Spannungsausbildung quergebohrter Wellen," *VDI Forschungsheft* 419, Vol. 14(b), 1943, Page 1.
11. S. Timoshenko—Reference 9, Pages 479-480.
12. J. O. Draffin and W. L. Collins—"Effect of Size and Type of Specimens on the Torsional Properties of Cast Iron," *Proc. ASTM*, Vol. 38, 1938, Page 235.
13. S. Timoshenko—Reference 9, Page 482.
14. C. R. Soderberg—"Working Stresses," *Handbook of Experimental Stress Analysis*, John Wiley & Sons, New York, 1950, Page 449.
15. R. E. Peterson and A. M. Wahl—Reference 3, Page A149.

DESIGN ABSTRACTS

Cushion Starting

. . . of a-c motors to prevent current peaks

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AS THE sizes of driving motors increase, more attention must be given to the effect of high starting currents on the power systems to which they are connected.

On power systems that have no automatic voltage regulation, a definite maximum current cannot be exceeded without causing such a voltage drop that other equipment may not function properly. On systems that have automatic adjustment of voltage (such as network systems), there is a limit to the increment of current at one time. As long as the current is increased in steps far enough apart to permit correction by the regulator, larger total currents can be accommodated.

This discussion is devoted entirely to the various means which may be used to "cushion" the shock on the line by reducing the starting current of induction motors. No consideration will be given to schemes which accomplish purely "torque" cushioning. The starters discussed will include, for squirrel-cage motors: autotransformer, primary resistance, primary reactance, star-delta, part winding, and electronic. Also, the secondary type for wound-rotor motors will be included.

AUTOTRANSFORMER STARTER: Referring to Fig. 1*a*, note that the autotransformer starter is so named because an autotransformer is connected to the line and the motor is connected to reduced voltage taps on it by means of a five-pole, full-current contactor. This contactor may

Table 1—Cushion Starters for Squirrel-Cage and Wound-Rotor Motors

Type Motor	Type Starter	Cost Index (Motor plus Starter)	Per Cent Full Voltage Values at Starting			Contactors		No. O. L. Relay	Auxiliary Equipment	Losses in Starting	Transition
			Terminal Voltage	Approx. Line Current	Torque lb-ft	No.	Per Cent Size				
Squirrel cage	Manual auto-transformer	100	50 65 80	30* 48* 71*	25 42 64	1-5P	100	2	Transformer	Low	Open
Squirrel cage	Magnetic auto-transformer	121	50 65 80	30* 48* 71*	25 42 64	1-5P 1-3P	100	2	Transformer	Low	Open
Squirrel cage	Magnetic primary resistance	119**	80	80	64	2-3P	100	2	Resistance	High	Closed
Squirrel cage	Magnetic primary reactance	121	50 65 80	50 65 80	25 42 64	2-3P	100	2	Reactance	Medium	Closed
Squirrel cage	Star delta	123		33	33	2-3P 1-2P	100	2		None	Open
Squirrel cage	Part winding	105		60	50	2-3P	50	4		None	Closed
Squirrel cage	Electronic										Closed
Wound rotor	Drum controller	193						2	Resistor	High	Closed
Wound rotor	Magnetic starter	220						2	Resistor	High	Closed

* Applies only to transformer designed for motor used. ** Same cost as auto-transformer above 40 hp, 440 v.

be one size smaller on the larger starters.

The simplest form of this starter is the manual type, in which the operator pulls the handle forward for starting, then pushes it back for running. It has the lowest cost of any of the starters, but it has some disadvantages. The time of changing from start to run is left to human judgment, and may result in unnecessary current peaks. Open transition is the standard arrangement. With this system there is a loss of torque during the transfer period because one contactor must open before the other closes to prevent short circuiting of the autotransformer.

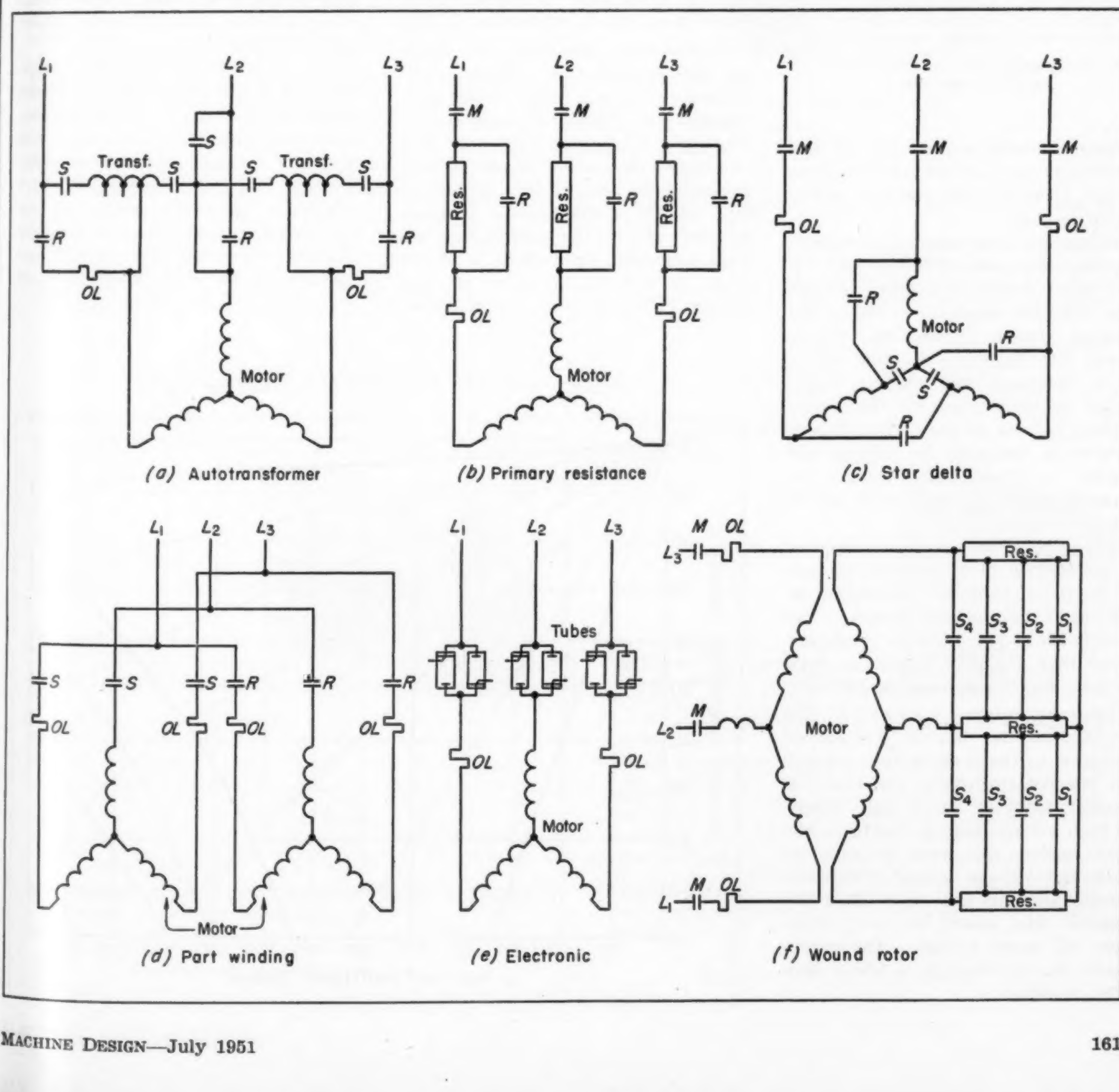
Taps usually are provided on the autotransformer so that 80, 65 or 50 per cent of the line voltage may be obtained. This results in starting torques of 64, 43 or 25 per cent of full-voltage torque. The selected percentage prevails during acceleration, giving a torque curve proportional to the original motor curve. The losses during starting are low. Therefore, line currents during starting are low, provided the transformer is designed for that size of motor. Since a transformer usually is designed for a range of motor horsepower ratings, not all currents will be the minimum possible.

Another form of this starter is the magnetic type, in which magnetic contactors are used to replace the hand operated ones already described, usually with time delay between the two steps. Automatic acceleration thus may be obtained with comparative simplicity. The magnetic form also lends itself to several variations, including methods of obtaining closed transition by the use of additional contactors.

A cost index, including a totally-enclosed, fan-cooled motor and each type of starter, is shown in TABLE 1. This index is an average of three typical motor sizes—25, 100 and 200 hp—for both 220 and 440 volts. Incidentally, it is interesting that the deviation from average is small for any one type of starter. With an index of 100 assigned to the manual type autotransformer starter, the simple magnetic autotransformer starter is 121.

PRIMARY-RESISTANCE STARTER: In this type of starter, shown in Fig. 1*b*, a resistor is connected in series with each phase, producing a reduced voltage at the motor terminals due to the IR drop of the starting current. When the motor has had time to accelerate, a time-delay relay and a contactor short out the resistors. This starter is simple, since only two 3-pole contactors, a timing relay and the resistor are required. It also has

Fig. 1—Circuit diagrams of various types of motor starters that can be used to cushion current shock for across-the-line starting



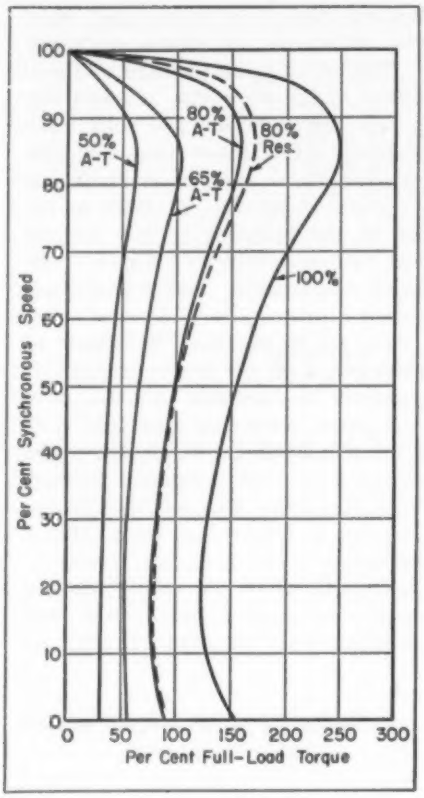


Fig. 2—Speed-torque characteristics of squirrel-cage motor

inherent "closed transition," as the resistor is shorted out and the line is not opened. The starting power factor is high.

It has the disadvantage of requiring more amperes of line current for the same available starting torque than the autotransformer, due to the resistor losses. Therefore, power losses are higher. As the motor speed increases, the current drops, which in turn increases the motor voltage and its torque. This characteristic is desirable for pumps and similar machines which require increased torque as they come up to full speed.

The commercial resistor is designed to produce 70 to 85 per cent voltage at the motor terminals, depending on the particular motor design. The cost index is 119, because prices are lower than the autotransformer type in the lower horsepower ranges only.

PRIMARY-REACTOR STARTER: The primary-reactor starter is almost identical to the resistor type, except for the substitution of reactors. A reactor, however, lends itself better to high voltage than a resistor. For that reason, this type is used on most high-voltage designs. The losses during starting are lower than the resistor type, which makes it desirable for large motors. The power factor during starting is lower than with resistors.

STAR-DELTA STARTER: The star-delta starter is used with a motor whose windings are delta-connected for running, connecting the windings in star for starting as illustrated in Fig. 1*c*. The voltage across each winding is reduced in the ratio of 1/1.732, and the line current is reduced to 1/3 of the delta connection. A fixed torque of approximately 1/3 of normal is obtainable. Two 3-pole and one 2-pole contactors are required—each full size—but no transformer, resistor, or reactor, and therefore no extra losses.

This starter lends itself well to motors of standard European connections such as 220/380-volt motors, but standard dual-voltage motors in this country are not designed for star-delta connections. They can, of course, be obtained on special order.

The torque curve for star connection follows the normal delta connection curve in general shape at 1/3 the normal values, and cannot be increased. Open transition is mandatory. The cost index, including the extra cost of special motor connections where applicable, is 123.

PART-WINDING STARTER: In a standard 220/440-volt, dual-voltage, star-connected motor, nine leads are brought out. These can be connected into two independent 220-volt star windings. If one of these is connected to the 220-volt line, as seen in Fig. 1*d*, the current inrush is about 60 per cent of the full-winding inrush, producing approximately 50 per cent of the full-winding starting torque. If the friction load and inertia are light enough, the motor will accelerate to nearly full speed. When the second winding is connected to the line, there will be a greatly reduced current inrush. Thus, the initial inrush would be the maximum current.

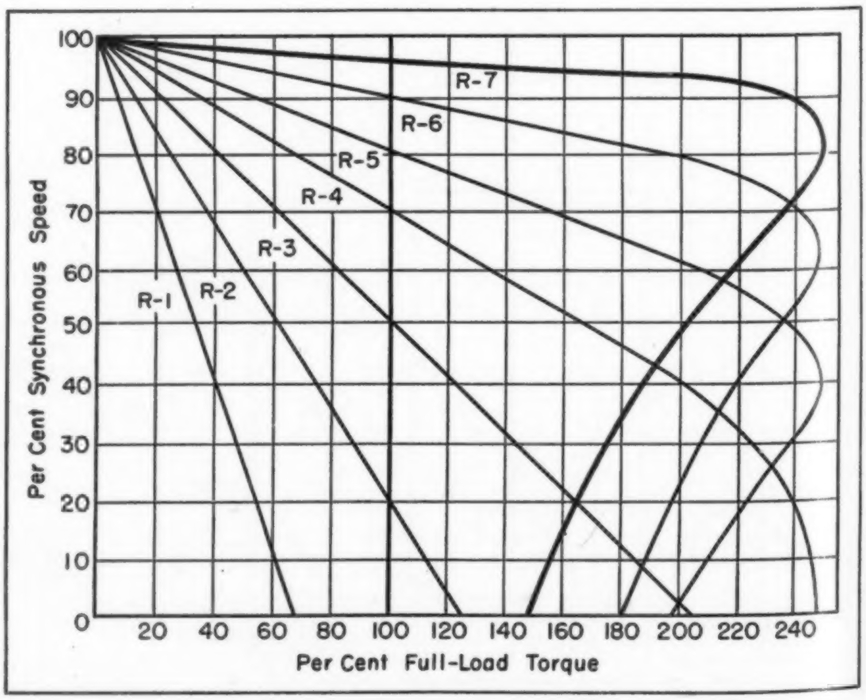
Time Delay Needed

If the load is high enough to prevent rotation, or to drag out acceleration for more than a few seconds, then the second winding must be connected—after a time delay. This will produce a second large inrush peak and increase the torque to normal. If the motor has not started at all, the locked rotor current then will be the same as if started on full winding, but such a characteristic still should be acceptable where increment starting is permitted. Two contactors, each rated one-half the normal horsepower and each with its own overload relay, are used for the two parts of the windings, together with a time relay. The torque would be on a curve approximately 50 per cent of the normal curve.

The leads brought out of a standard, dual-voltage, delta-connected motor can be used if one of the contactors is made full horsepower, but it is not ordinarily suitable for connecting into two independent part windings. However, six leads for part windings easily can be brought out. Single-voltage motors and motors for 440 and 550-volt service must

(*Continued on Page* 200)

Fig. 3—Speed-torque characteristics of wound-rotor motor

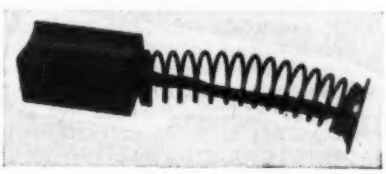


NEW PARTS AND MATERIALS

. . . presented in quick-reference data sheet form for the convenience of the reader. For additional information on these new developments, see Page 179

Motor Brushes 1

Pure Carbon Co. Inc., St. Marys, Pa.



Style: Split or sandwich; with pigtail and spring

Size: To specifications

Service: Commutation on motors and generators operated at high altitudes; for close control of protective film on commutator or ring, exact amount of chemical is contained in each brush

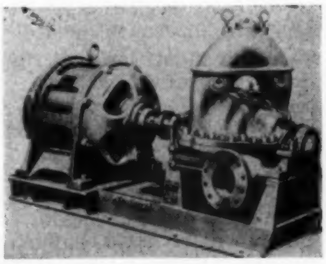
Design: Chemically treated section sandwiched between untreated layers of one-half to one-third the thickness for altitudes to 50,000 ft; split construction on brushes ⅜-in. thick or greater where equal parts can be used; combination of treated and untreated bruches where two or more small brushes are used in tandem

Application: Aircraft motors and generators.

For more data circle MD 1, Page 179

High Head Pump 2

Economy Pumps Inc. Div., Hamilton-Thomas Corp., Hamilton, O.



Designation: Type DMD

Style: Two-stage, opposed impeller; single inlet.

Size: 2 in. to 10 in. discharge

Service: Capacities to 4000 gpm; heads to 750 ft.

Design: Horizontally split casing with interconnecting passage between two opposed impellers; impellers hydraulically and dynamically balanced; dust-proof cartridge mounted ball bearings, angular contact duplex type for thrust, double row for radial loads; metal step-type water flingers shield bearings; alloy steel shaft

Application: Boiler feeds, chemical plants, hydraulic elevators, oil refineries, water works.

For more data circle MD 2, Page 179

Brazing Rings 3

Lucas-Milhaupt Engineering Co., 5051 S. Lake Dr., Cudahy, Wis.

Designation: No-Tangle

Style: Flat wire rings; gap, butt or lap

Size: To specifications

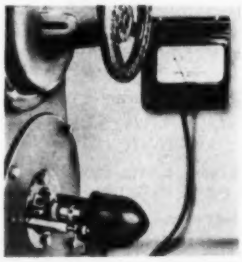
Service: Silver brazing or soldering where flattened shape and large volume of alloy is needed; stress relieved to retain ±0.001 in. tolerance; will not distort or fall away from work piece when heat is applied

Design: Preformed of flat silver alloy wire in notched coil; available in Sil-Flos and Easy-Flo wire; gap rings for inside applications, lap and butt for close-tolerance applications on keyed or nonkeyed outer surfaces.

For more data circle MD 3, Page 179

Electric Tachometer 4

Reeves Pulley Co., Columbus, Ind.



Style: A-c generator principle; furnished with mounting for Reeves variable speed transmissions.

Size: Mounting plate width 5 to 8⅜ in.; overall depth 5$\frac{1}{16}$ to 6 in.

Service: For registering rpm—may be used for time, fpm, lb/hr, etc.; direct coupled when low speed of shaft above 100 rpm and high speed of shaft above 750 rpm, gear coupled otherwise; indicator head mounted up to 300 ft from generator unit.

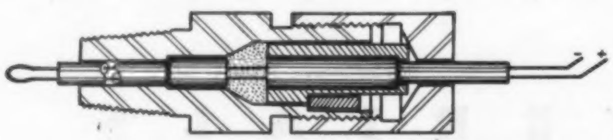
Design: Indicator head is standard voltmeter calibrated in rpm or other units if desired; a-c generator coupled to shaft supplies variable voltage; ten ft of two-conductor cable supplied; zero-set screw on indicator.

For more data circle MD 4, Page 179

NEW PARTS AND MATERIALS

Thermocouple Gland 5

Conax Sales Co. Inc., 4515 *Main St., Buffalo* 21, *N. Y.*



Style: Bare wire; ceramic and compressed talc sealed

Size: For 20 gage, 14 gage thermocouple wires

Service: Sealing bare-wire thermocouples in pressure vessels; temperature range −100 to +1600 F; pressures to 4000 psi, or vacuum; terminal block or conduit connections

Design: Wires sealed by compressing dry powdered talc between two ceramic insulators; insulators slid into place on wire, placed in container fitting with powdered talc; tool-steel gland follower and key placed in position and hex cap nut tightened, compressing talc; cap nut and fitting, type 304 stainless steel.

For more data circle MD 5, Page 179

Miniature Tube Sockets 7

Teflon Products Div., United States Gasket Co., P. O. Box 93, *Camden, N. J.*

Designation: Chemelec SO 428.

Style: 7-pin miniature.

Size: In accordance with type TS7T101 of JAN S-28A or R.M.A. requirements.

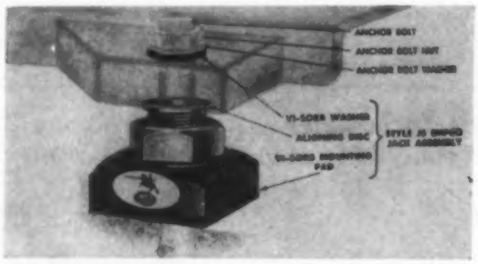
Service: For high or low ambient temperature operation or maintaining high frequency stability; loss factor 0.0005; dielectric constant 2.0 from 60 cycles to 30,000 megacycles; temperatures from −110 to 575 F with negligible change in dielectric strength and power factor; noncarbonizing under arcing; water absorption rating of 0.0% by ASTM test; chemically inert, not affected by extreme humidities, corrosive atmospheres or fungus.

Design: Assembly of Teflon socket with mounting and electrical contact hardware; other mounting accessories available.

For more data circle MD 7, Page 179

Machine Leveling Jack 6

Enterprise Machine Parts Corp., 2731 *Jerome Ave., Detroit* 12, *Mich.*



Designation: Empco JS

Style: Jack base, nut, lift screw and aligning disk assembly

Size: ⅜-in. leveling adjustment; maximum height 2⅛, 2⅜, 3⅜ in; clearance for ¾-in. diam anchor bolt

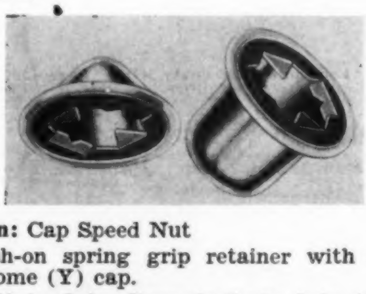
Service: Supports 20 tons; lifts 8000 lb; concave lift screw top fitting into convex spherical radius on bottom of aligning disk compensates for uneven floors and permits sufficient float for level alignment of machine

Design: Hollow jack for support at anchor bolts; turning adjusting nut with wrench raises or lowers lift screw; vibration absorbent mounting pads ¼ to 1 in. thick available.

For more data circle MD 6, Page 179

Decorative Retaining Nuts 8

Tinnerman Products Inc., P. O. Box 6688, *Dept.* 14, *Cleveland* 1, *O.*



Designation: Cap Speed Nut

Style: Push-on spring grip retainer with cylindrical (Z) or dome (Y) cap.

Size: For ⅛ to $\frac{7}{16}$-in. diam shafts in $\frac{1}{16}$-in. increments; type Y, ⅞-in. diam $\frac{21}{32}$-in. height; type Z, $\frac{23}{32}$-in. diam, $\frac{31}{32}$-in. height.

Service: Light applications on exposed shaft ends; decorative hub cap and wheel retaining duty; push-on application; rotating or stationary shaft ends; effective on metal or plastic.

Design: Assembly of aluminum cap and round spring-steel retainer having 6-pointed grip hole; type Y has rounded top for short shaft ends, type Z has longer cylindrical extension for greater shaft protrusions; available in many colors and finishes.

Application: Coaster wagons, scooters, tricycles, supermarket carts, mobile toys.

For more data circle MD 8, Page 179

JOHNSON BRONZE

SLEEVE BEARING DATA

Bearing TYPES

JOHNSON BRONZE

SLEEVE BEARING DATA

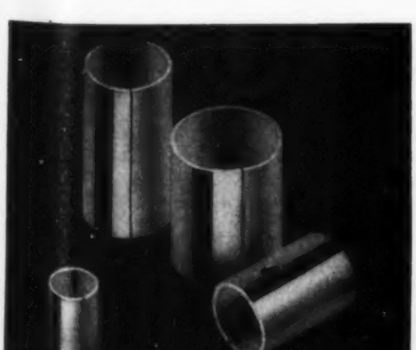
Sheet Metal Bearings-3

Steel-Back Babbitt-Lined Bearings

THIS TYPE of Sleeve Bearing is usually referred to as "laminated". This because two separate metals are firmly bonded together. In the automotive industry, the largest user, they are sometimes referred to as "Slip-in" bearings.

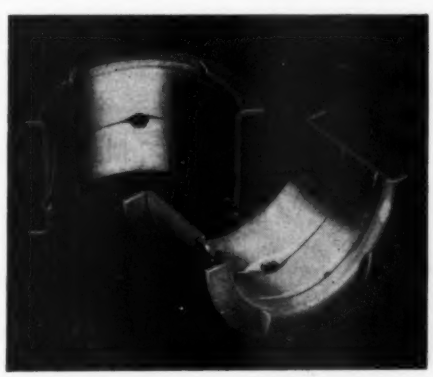
The first step in the manufacturing process is the coating of the steel strip—S.A.E. 1010—with either tin-base or lead-base babbitt . . . as the application may require. This is a continuous process with exact temperature control and necessary precautions to prevent oxidation. Extreme care must be exercised in this operation in order to secure a perfect, durable and lasting bond of the two metals. Next, the strip is brought to the desired thickness. The balance of the manufacturing procedure is essentially the same as in other types of sheet metal bearings . . . stamping, forming and finishing to precise size. Steel and babbitt bearings are available in a wide range of sizes, plain or flanged, half bearings or cylindrical. Oil grooves, slots or holes can be incorporated during stamping operation.

Several standard thicknesses of steel are used, depending on the nature of the application. For example: .036, .050, .060, .070, .080, .090 and .100. In general applications the thickness of the babbitt will run from .015 to .020. While on precision-type bearings it is held to .002 to .005.



The selection of the babbitt depends quite naturally on the bearing application. Lead-base babbitt is much more economical than tin-base and in certain alloys will provide a higher load carrying capacity. While tin-base is more expensive it has better corrosion resistant properties and a higher fatigue strength. Both materials have performed very well in automotive, diesel and electric motor applications, with a slight preference for tin-base babbitt in main and connecting rod bearings.

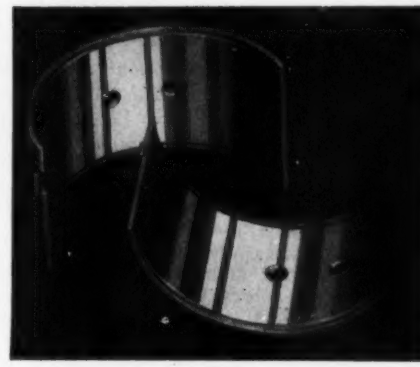
While steel-backed babbitt-lined bearings are best suited for light to moderate loads with high speed, there is a definite relation between the thickness of the babbitt and the



load carrying capacity of the bearing. A babbitt thickness of .020 has a generally rated load capacity of 1500 pounds per square inch whereas by holding the babbitt to .002 to .007 increases the capacity up to 2000 pounds per square inch. This results in longer bearing life.

When babbitt-lined bearings are used within the range of their load capacity they are generally considered the best bearings available. They can be used with steel shafts of practically any hardness. It must be remembered, however, that as the hardness of the shaft increases, the resistance to wear increases.

The method of installing steel and babbitt is practically the same as bronze and steel or bronze sheet metal. We recommend that they be



line reamed or bored in assembly. Burnishing should be avoided if possible.

Main bearings are usually produced on specially designed equipment to keep within a tolerance of .00025 and .0005. This accuracy permits any two halves to make one complete bearing, thus simplifying installation.

Some of the most desirable qualities of steel and babbitt bearings are:

1. Good wettability with lubricants.
2. High corrosion resistance to most oils at operating temperatures.
3. High embeddability.
4. Good conformability.
5. Work well with soft shafts.

•

Engineering Service

Johnson Bronze offers manufacturers of all types of equipment a complete engineering and metallurgical service. We can help you determine the exact type of bearing that will give you the greatest amount of service for the longest period of time. We can show you how to design your bearings so that they can be produced in the most economical manner. As we manufacture all types of Sleeve Bearings, we base all of our recommendations on facts free from prejudice. Why not take full advantage of this free service?

This bearing data sheet is but one of a series. You can get the complete set by writing to—

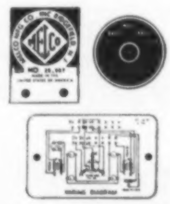
Johnson Bronze

SLEEVE BEARING HEADQUARTERS
525 S. MILL ST. • NEW CASTLE, PENNA.

NEW PARTS AND MATERIALS

Nameplates 9

Duralith Corp., 136 *N. Seventh St., Philadelphia* 6, *Pa.*



Style: Laminated thermosetting plastic

Size: Up to 11½ x 11½ in., thickness greater than 1/32-in.

Service: For dials, signs, markers, instruction panels, diagrams; heat acid, alkali, solvent, moisture resistant duty; conform to military specifications MIL-P-78, type GCP-H; nonglare surface; for flat surfaces only.

Design: Melamine or phenolic core laminated on both sides with melamine overlays; printing with special inks in colors desired on laminating paper, covered by 0.003 in. of transparent melamine overlay, cured under heat and pressure to obtain hard surface; specials in translucent material for back lighting, transparent acrylic adhered to melamine nameplates for edge lighting, phosphorescent and fluorescent.

Application: Machine tools; control panels; electrical controllers; circuit breakers; aircraft.

For more data circle MD 9, Page 179

Variable Speed Pulley 11

Gerbing Mfg. Co., 11801 *Milwaukee Ave., Northbrook, Ill.*

Designation: Roto-Cone 1300-150

Style: Self adjusting variable pitch.

Size: 13 in. diam sheave with 30 degree included angle groove; 12 9/16 in. overall length with 4½ in. diam hub; 1.250 in. or 1.625 in bore; weight 59 lb.

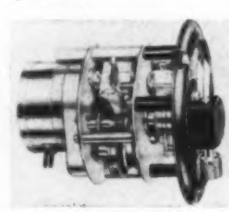
Service: 3:1 ratio—12.313 in. PD max to 4.104 in. PD min; 15 hp at 1750 rpm, 10 hp at 1150 rpm, 7.5 hp at 8660 rpm; horsepower rating decreases proportionally to reduction in driven speed; for standard Vari-speed rubber V-belt.

Design: Rack and gear arrangement controls sheave movement and maintains belt alignment at all speeds; dynamically and statically balanced.

For more data circle MD 11, Page 179

Time Delay Relay 10

R. W. Cramer Co. Inc., Centerbrook, Conn.



Designation: TEC, TER

Style: Automatic or controlled reset; differential planetary gear clutch, electromagnetically operated.

Size: Flush panel mounting, 3¾ in. diam x 4 13/16 in. deep; conduit box, 4 31/32 in. square x 6 15/16 in. deep; dustproof conduit connection box, 7 11/16 in. high x 5¼ in. wide x 6 in. deep.

Service: For adjustable or fixed time delay between operation of control circuit and subsequent closing or opening of load circuit; 15, 30, 60 sec, 2, 5, 15, 30, 60 min, 3 and 6 hr time ranges; nominal switch rating 10 amp at 115 v, 5 amp at 230 v; 50-60 cycle a-c operation, specials for 115 v, 25 cycle a-c; in TEC, timer resets when control circuit is de-energized or power fails; in TER, time cycle is resumed at same point after power failure, momentary energizing of clutch brake relay resets timer.

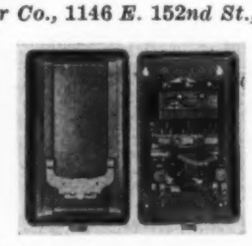
Design: Synchronous motor operates through sun gear and planet gears to output; brake relay locks internal gear to drive output when energized on TEC, when de-energized on TER; micrometer adjusting knob; die-cast front plate; Bakelite sleeve and rear plate.

Application: Plastics molding, heat treatment, lapping, grinding, induction heating, mixing, agitating, ovens, and other machines.

For more data circle MD 10, Page 179

A-C Motor Starter 12

Clark Controller Co., 1146 *E. 152nd St., Cleveland* 10, *O.*



Designation: CY-2

Style: Twin-break magnetic with magnetic blowout; 3-wire.

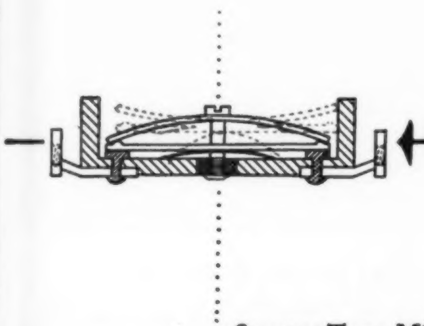
Size and Service: NEMA size 2; full voltage starting with thermal overload protection; burning and pitting of contact minimized by arc moving constantly over face of contact; noncarbonizing; phase to phase failure caused by accumulation of ionized gases between wiring terminals prevented by closed top of arc chamber.

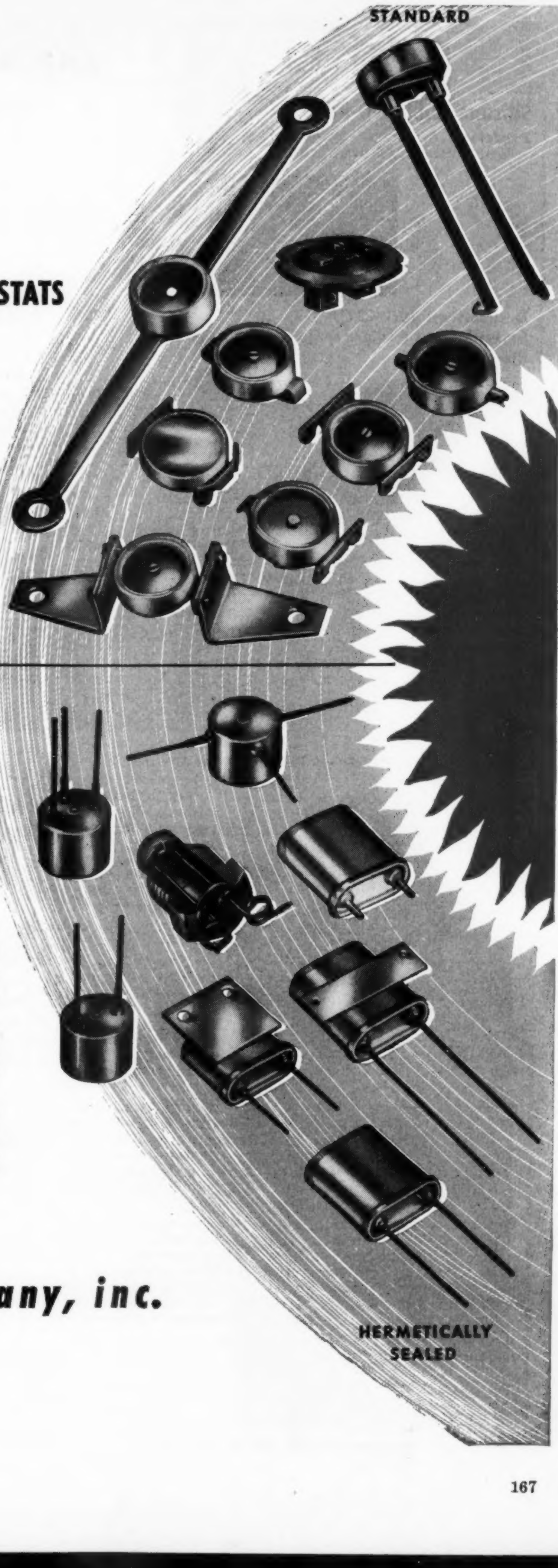
Design: NEMA standard enclosures, mounting dimensions and wiring; vertical lift clapper type magnet; silver alloy power and control contacts; movable contacts, axially aligned with stationary contacts, make contact with straight line motion; blowout coil, concentric with and above upper stationary contact, forms cone shaped magnetic field; in one half cycle, arc is forced to outer edge of upper contact, lengthened and rotated; other half cycle rotates arc and forces it to middle; both actions tend to quench arc; steel arc chambers; melting alloy type overload relays; two normally open or closed control circuit contacts can be added.

For more data circle MD 12, Page 179

For quick make and break *specify* STEVENS TYPE M THERMOSTATS

- APPLIANCES
- AIR CONTROL EQUIPMENT
- ELECTRONIC DEVICES
- AVIONIC EQUIPMENT
- THERMAL TIMERS
- INSTRUMENTS





Stevens Type M* thermostats are engineered for compactness . . . lightness . . . close temperature control. Featuring *quick make and break,* fast snap of bimetal disc and double series contacts reduce arcing . . . assure positive On and Off.

Bimetal rests on either a monel-backed or a nickel silver-backed contact disc which carries current. Electrically independent bimetal eliminates artificial cycling and life-shortening "jitters."

Supplied with virtually any type terminal in standard or hermetically sealed styles, Type M thermostats give stable operation in ambients from -75°F to 600°F.

Get faster response . . . closer temperature control. Specify Stevens Type M thermostats in your product—*for better performance, longer life.*

*PATENT APPLIED FOR

STEVENS

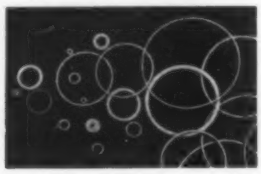
manufacturing company, inc.

MANSFIELD, OHIO

NEW PARTS AND MATERIALS

Silicone Rubber O-Rings 13

Frederick S. Bacon Laboratories, 192 *Pleasant St., Watertown* 72, *Mass.*



Designation: 407B-217-1

Style: Molded toroidal shape

Size:

Part Letter	Size (ID x diam section, in.)	Part Letter	Size (ID x diam section, in.)
U	0.094 x 0.057	P	0.375 x 0.070
K	0.094 x 0.068	T	$1\frac{11}{16}$ x 0.070
X	0.219 x 0.070	S	$2\frac{3}{8}$ x 0.070
L	0.250 x 0.057	N	$2\frac{11}{16}$ x 0.050
M	0.250 x 0.059	J	$2\frac{11}{16}$ x 0.070
Y	0.250 x 0.070	C	$2\frac{7}{8}$ x 0.063

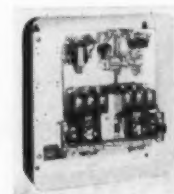
Service: High temperature static shaft sealing; tensile strength at break, greater than 500 psi; elongation at break, greater than 80; hardness (Shore A 2), 70-80; compression set at 300 F (ASTM 395-46T, method B), less than 33%; brittle point, less than −70 F; swell in representative fluorocarbons, chlorinated biphenyls and petroleum oils, less than 6%

Design: Red color; special sizes and shapes in rubbers with other properties available; tolerances on dimensions—U, K, X, L, M, Y, P ±0.005 in. on ID, ±$\frac{1}{32}$-in. on ID for T, S, N, J, C—on section, ±0.002 in. for U, K, ±0.003 in. for X, L, M, Y, P, T, S and ±0.005 in. for N, J, C.

For more data circle MD 13, Page 179

Motor Reversing Control 15

Allen-Bradley Co., Milwaukee 4, *Wis.*



Designation: Bulletin 723

Style: Package automatic cycle control; two or three-wire; panel type

Size: Sizes 00, 0, 1—$14\frac{5}{8}$ in. high x $12\frac{3}{8}$ in. wide x $4\frac{3}{16}$ in. deep, size 2—$22\frac{1}{4}$ in. high x $15\frac{5}{8}$ in. wide x $5\frac{3}{16}$ in. deep

Service: Reversing drive with dwell; for squirrel cage motors, 50-60 cycle, 3 phase a-c; four reversals per minute, dwell adjustable from $\frac{1}{4}$ to 10 sec, specials for 2 or 3 reversals per min; size 00, 110 v—$\frac{3}{4}$-hp, 220-440-550 v—1 hp; size 0, 110 v—$1\frac{1}{2}$ hp, 220-440-550 v—2 hp; size 1, 110 v—3 hp, 220 v—5 hp, 440-550 v—$7\frac{1}{2}$ hp, size 2, 110 v—$7\frac{1}{2}$ hp, 220 v—15 hp, 440-550 v—25 hp.

Design: Two or three-wire control for maintained or momentary contact pushbutton operation, respectively, size 2 for three wire control only; overload relay; 25 cycles available; enclosures for open, general-purpose (NEMA I), or Class I, Group D (NEMA VII) hazardous locations

Application: Laundry and dry cleaner washing machines.

For more data circle MD 15, Page 179

Wire Resistor 14

Mullenbach Electrical Mfg. Co., 2300 *E. 27th St., Los Angeles* 58, *Calif.*

Style: Open coil wound; open or enclosed bank assemblies

Size: $13\frac{11}{16}$ in. long x $3\frac{5}{8}$ in. wide x 1 in. thick; shipping weight $1\frac{1}{2}$ lb.

Service: Nineteen continuous-duty ratings from 0.25-ohm at 60 amp to 20 ohm at 5 amp, 500 v a-c or d-c; ratings based on 375 C temperature rise; special alloy resistance wire gives constant resistance and resists corrosion at all working temperatures; UL approved; complies with USN Bur. Ships Spec. 17-P-4 Type EW 150 ft-lb shockproof

Design: One-piece stamped cold rolled steel frame, cadmium plated; operates at ground potential; porcelain insulators grooved to provide proper wire spacing; two thicknesses of mica-fiberglass-silicone insulation separate porcelain insulators from frame and cushion porcelains from shock; one terminal each end brazed to wire, extra tap terminals if desired

Application: Battery charging, motor control, load banks, arc lamp ballasts.

For more data circle MD 14, Page 179

Integrator 16

Librascope Inc., 1607 *Flower St., Glendale* 1, *Calif.*

Style: Ball and disk

Size: 3.094 in. high x 2.750 in. wide x 1.875 in. deep; disk drive shaft 0.3744/0.3746-in. diam x 0.725-in. extension; ball carriage displacement shaft 0.1869/0.1871-in. diam x 0.531-in. extension each side plus 1.50 in. travel extension either side; cylinder shaft 0.2494/0.2496-in. diam x 0.531 extension each side; weight 21 oz

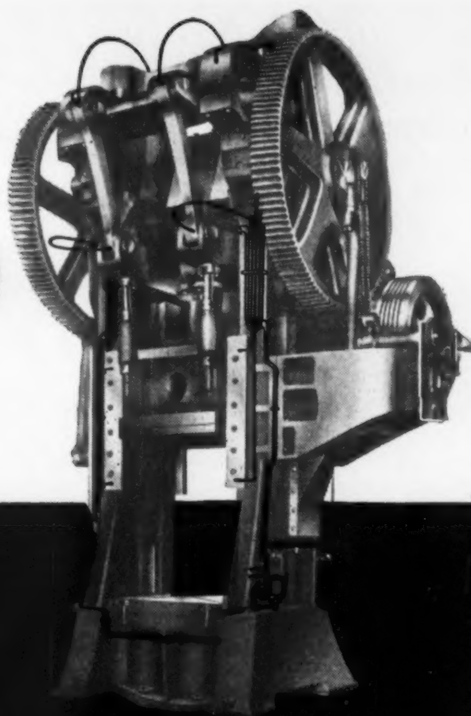
Service: Generating derivatives, integrals, reciprocals, products, squares, exponentials, trigonometric functions; smoothing or averaging fluctuating data; 0.8 in-oz max input torque at zero load; 1 oz force to move ball carriage; precision (reproducibility) 0.01%; max disk speed 750 rpm

Design: Constant speed rotation of horizontal lapped carboloy disk rotates balls which rotate cylinder; ball carriage displacement shaft moves balls radially with respect to disk and axially with respect to cylinder to vary rpm of cylinder output shaft; anodized aluminum alloy case; other exposed parts stainless steel

Application: Industrial computing systems, analog computers, military fire control systems, controls for closed loop servo systems.

For more data circle MD 16, Page 179

How To Design Extra Production Time into Every Machine



ALEMITE Accumeter automatic lubrication

reduces maintenance, increases machine output

You add extra production time to your machines by removing the need to shut down for lubrication . . . when you *design in* Alemite Accumeter Automatic Lubrication! Consisting of a lubricant pump, a distribution system of tubing, and *force-feed* valves for individual bearings—an Accumeter System makes lubrication foolproof. It automatically lubricates every bearing on a machine . . . from one central point . . . *while production continues*. Ends the risks of errors or neglect by your customers. Assures greater production, lower maintenance costs, longer machine life!

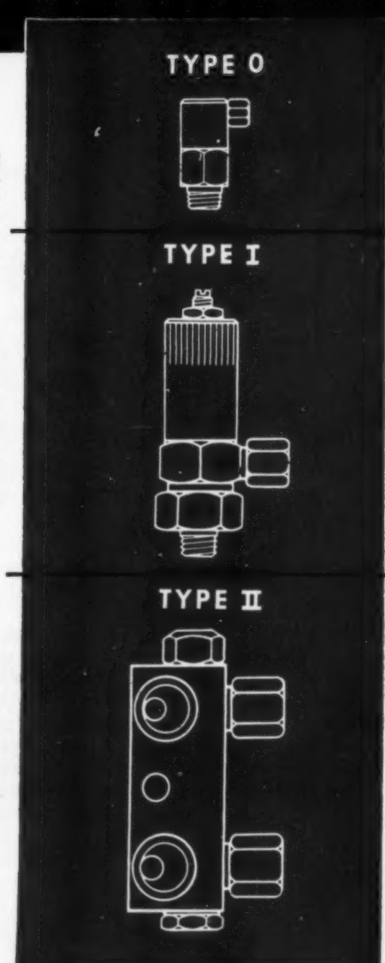
No other method has equaled Alemite's Accumeter Systems for sustained accuracy in metering oil or grease to bearings. Tests show *no variation* in the amount of lubricant discharged to bearings . . . even after 73,312 lubrication cycles, equal to 122 years of twice-a-day service! Moreover, lubrication is either fully hydraulic or continuous between cycles, thanks to the exclusive Alemite "accumulating" feature that *prolongs* the discharge of lubricant to bearings!

To cover your full range of requirements, there are three different Alemite Accumeter Automatic Lubrication Systems. Versatile in application, they are adaptable to virtually any machine. Send now for free bulletin giving full data. Alemite, Dept. R-71, 1850 Diversey Pkwy., Chicago 14, Illinois.

MIDGET OIL VALVES
Fixed output metering valves for single line system serving up to 200 small bearings. Especially suited to precision machines, and wherever space is limited.

ONE-BEARING VALVES
Oil or grease metering valves, with *fixed or adjustable output*, for single line system. Valves for any bearing capacity. Single system serves up to 400 bearings.

TWO-BEARING VALVES
Adjustable output metering valves for oil or grease. Each serves 2 bearings. Valves fully sealed, hydraulically operated on both load and discharge cycles. System handles up to 600 bearings. Manual or automatic operation.



ALEMITE

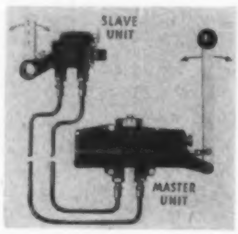
1850 Diversey Parkway, Chicago 14, Illinois

Another Product of STEWART WARNER

NEW PARTS AND MATERIALS

Hydraulic Remote Control — 17

Superdraulic Corp., 14256 Wyoming Ave., Detroit 4, Mich.



Style: Synchronized master and slave hydraulic units.

Size: To specifications.

Service: For straight line or arc segment control movement; 500 in-lb torque; positive load carrying ability in either direction; automatic lock in slave lever provides irreversibility, available without locking feature; either unit mounted stationary or movable in any position.

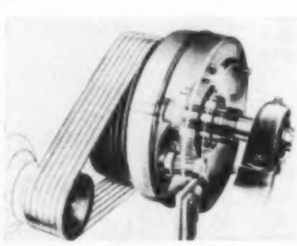
Design: Motion applied to master unit actuating lever duplicated by slave unit lever; control system provides for expansion and contraction of both fluid and metal due to temperature changes; self-lubricating.

Application: Machine tools, test stands, power plants, textile, food and processing machinery, road building, lumbering and farm machinery, assembly fixtures, ships, aircraft controls.

For more data circle MD 17, Page 179

Reduction Drive — 18

American Pulley Co., 4200 Wissahickon Ave., Philadelphia 29, Pa.



Designation: No. 6

Style: Shaft mounted; horizontal or vertical; helical gear driven; torque arm

Size: 40 hp; 23½ in. diam, 13$\frac{3}{16}$ width; interchangeable bushings adapt female output hub to various shaft sizes from 2$\frac{7}{16}$ in. to 3$\frac{15}{16}$ in.; input shaft 2$\frac{7}{16}$ in. diam with 4$\frac{11}{16}$ extension

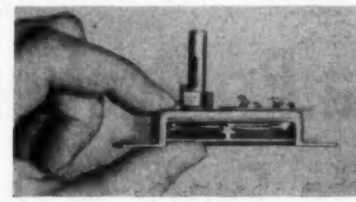
Service: 13.19:1 reduction; ratings range from 6 hp at 11 rpm to 42.8 hp at 110 rpm; overload protection with special torque-arm overload release; input pulley on reducer 12 in. min.

Design: Unit mounts on machine input shaft; forged, heat-treated alloy steel helical gears; internal ribbed cast iron housing for rigidity; anti-friction bearings; oil seal located above oil level to prevent leakage.

For more data circle MD 18, Page 179

Bimetal Thermostat — 19

Stevens Mfg. Co. Inc., 69 S. Walnut St., Mansfield, O.



Designation: Type W

Style: Electrically independent bimetal; snap action make or break; adjustable or nonadjustable

Size: Adjustable—3.092 in. long x ⅝-in. wide x $\frac{23}{32}$-in. high with adjusting stem height extension as required; nonadjustable—3.092 in. long x ⅝-in. wide x $\frac{23}{32}$-in. high with $\frac{25}{32}$-in. stem extension

Service: Temperature control up to 600 F with 5 F fixed differentials up to 400 F, slightly greater up to 600 F; rated at 12 amp, 115 v a-c or 8 amp, 230 v a-c, no d-c rating; electrically independent thermostat eliminates artificial cycling; no interference with radio reception

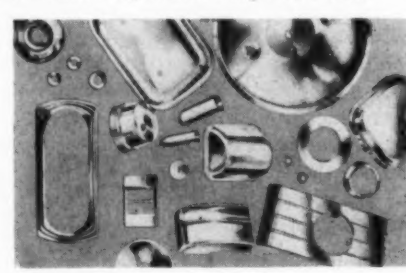
Design: Thermal element curves upward to force stainless steel crimped "snapping link" against pressure spring; when predetermined point is reached, spring pressure snaps link downward, breaking electrical contact; angular rotation of adjusting stems up to 300 degrees; nonadjustable type preset and sealed; silver contacts

Application: Rectifier fans, electronic devices, precise control of high wattage heater loads, refrigerator butter warmers, other appliances.

For more data circle MD 19, Page 179

Brass Alloy — 20

American Brass Co., Waterbury 20, Conn.



Designation: Formbrite.

Form: Sheet, strip, wire, rod, tube.

Service: For deep drawing, forming, cold heading or upset, cold drawing, machining, threading.

Properties: Fine grain size (about 0.008 mm) due to procedure of rolling or drawing and annealing brass containing 63 to 90% copper; initial high strength and stiffness; good polishing and finishing characteristics; resists abrasion and scratching; physical properties (70/30 Formbrite brass)—yield point (0.5% extension under load) 39,000 psi, tensile strength 63,000 psi, elongation in 2 in. 35%.

Application: Stamped, formed, drawn machine parts; cold heading rivets, machine screws, seamless tubing.

For more data circle MD 20, Page 179

For . . .

Gear Reduction Units
Aircraft Reciprocating Engines
Automotive Accessories
Jet Propulsion Units
Washing Machines
Standard & Special Machine Tools
Electrical Power Equipment
Business Machines

If *you* have a shaft sealing problem, Gits experience in these and many other specific applications can prove of great and immediate value to you.

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*Cartridge Seal . . . pressure balanced . . . requiring only 25% more space than lip-type seals.

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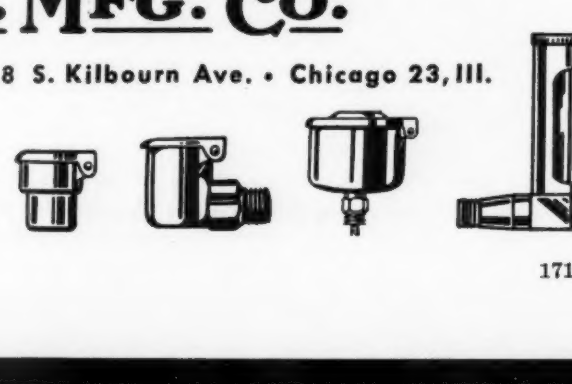
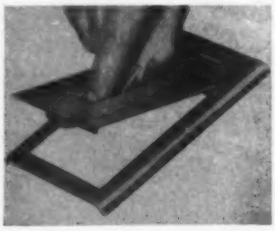


Flush Latch 21

Hartwell Co., 9035 Venice Blvd., Los Angeles 34, Calif.



Designation: H-4000

Style: Self-closing; flush handle.

Size: 1.850 in. long with 0.553-in. bolt extension x 1.502 in. wide x 0.576-in. thick.

Service: Interior or flush exterior installation on sheet metal, plastics, plywood or solid wood; nonself-closing if desired.

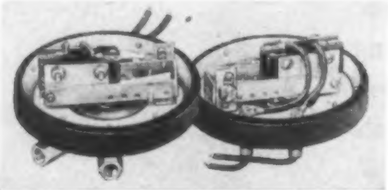
Design: Pressing down on front end of flush handle attached to latch trigger retracts bolt and lifts back end of handle for opening door or access cover; handle of any material, spot welded, riveted or screwed to trigger; pushbutton operation possible; adjustment for door thickness with spacers under handle; corrosion-resistant steel, aluminum alloy or cadmium-plated steel, 0.050-in. thick; music wire spring can be varied in design to withstand given load or vibration conditions.

Application: Inspection and access doors, removable panels, cabinet doors and drawers, cleanout plates, service doors.

For more data circle MD 21, Page 179

Pressure-Actuated Switch 22

F. W. Dwyer Mfg. Co., 317 S. Western Ave., Chicago 12, Ill.



Designation: Model 1620, Model 1625

Style: Diaphragm actuated micro type switch; front, flush or semiflush panel mount.

Size: 1620—4½ in. diam, 1625—7 in. diam; ⅛-in. pipe or rubber tube connections.

Service: Switch, 10 amp, 110 v a-c; plus, minus or differential pressures up to 4 in. water; model 1620 has minimum difference between make and break pressures of 0.20 in. water, minimum setting of 0.25 in. water; model 1625 has minimum make and break difference of 0.05 in. in lower ranges, 10% of setting in higher ranges; settings maintained within ±3%.

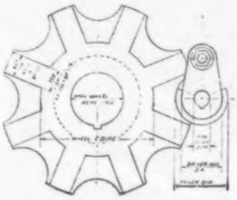
Design: Slack diaphragm of switch is linked to beryllium copper spring; pressure changes actuate spring which opens or closes micro type switch; settings and gap adjustable; mounts any position.

Application: Indicating pressure drops in air filters or across coils; indicating power or blower failure; actuating motors on automatic air equipment; with pitot static head as air velocity limit indicator; indicating pressure differences in forced air feed and boiler drafts.

For more data circle MD 22, Page 179

Geneva Drive 23

Geneva Machine & Tool Corp., 402 Ellamae Ave., Tampa, Fla.



Style: External closed Geneva.

Size: Minimum wheel bore ⅞-in. diam; max wheel bore 1½ in. diam, except 4-point ⅞ to 1½ in. diam; max wheel hub projection 2.250 in; center distances from 3 to 5¾ in.

Type	Wheel Diameter (in.)	Driving Radius (in.)	Driver Bore (max diam in.)
4-point	4.243 to 8.131	2.1213 to 4.0658	1 to 1½
5-point	4.854 to 9.303	1.7633 to 3.3797	1 to 1½
6-point	5.196 to 9.959	1.500 to 2.875	¾ to 1
8-point	5.543 to 10.624	1.148 to 2.2004	7/16 to ⅝

Service: Indexing to 4, 5, 6 or 8 positions in 360 deg; indexing to other positions with Geneva drive plus auxiliary gears.

Design: Cast iron; needle bearing driving pin.

For more data circle MD 23, Page 179

Manual Starter 24

Square D Co., 4041 North Richards St., Milwaukee 12, Wis.

Designation: Class 2510, Type A

Style: Fractional horsepower; single or double pole; thermal overload relay; neon pilot light if desired.

Size and Service: For 115 and 230 v a-c single phase and d-c motors; single-pole rating—1 hp for a-c, ½-hp for d-c; double-pole rating—1 hp for both a-c and d-c; contacts cannot be manually held closed against overload;

Type	Service	Size (height x width x depth, in.)
AO	open	3⅞ x 1 11/16 x 1 17/32
AG	NEMA I general-purpose	4 7/16 x 2⅝ x 2 5/32
AW	NEMA IV, V water and dusttight	5 5/32 x 2 13/16 x 4 3/32*
AR	NEMA VII, IX Class I, Group D Class II, Group E, F, G hazardous locations	6 11/16 x 3½ x 4 5/16

*5 1/32 x 6½ x 5 5/32 with pilot light.

Design: Double break contacts of fine silver; operating handle moves to midway position on overload; relay units interchangeable from front.

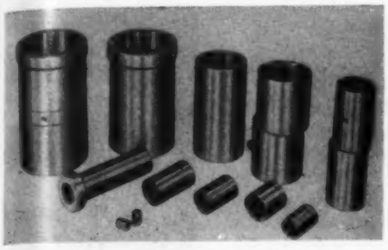
Application: Small machine tools, fans, pumps, oil burners, blowers, unit heaters or general use.

For more data circle MD 24, Page 179

NEW PARTS

Bronze Bearings 25

Bronze Bearings Inc., 544 North Ave. E., Cranford, N. J.



Style: Plain, flanged, split

Size: Standard OD ¾ to 10 in., length to 13 in; specials to 30 in. OD x 24 in. long or 22 in. OD x 30 in. long; weight of individual unit not to exceed 800 lb.

Service: Composition of cast bearing metal, and lubrication design to requirements

Design: Finishes range from semifinish to accurate machine finish; flanged types can be built with special flanges.

For more data circle MD 25, Page 179

Alkyd Plastic 26

Plaskon Div., Libbey-Owens-Ford Glass Co., 2112 Sylvan Ave., Toledo 6, O.

Designation: Plaskon alkyd 422

Form: Mineral filled; furnished in granular form

Service: High-speed automatic compression molding; electrical insulation; heat and arc resistance; dimensional stability

Properties: Thermosetting; self-extinguishing; water absorbtion (ASTM D570) 0.05-0.08%; arc resistance (ASTM D495) 180 + sec; dielectric strength 420 v per mil; dielectric constant 60 cycles 5.8-6.0, 1000 kc 4.2-4.3; dissipation factor 60 cycles 0.035-0.040, 1000 kc 0.014-0.015; insulation resistance 15 days at 70 C and 96% RH, 1000-1500 megohm; specific gravity 2.20-2.23; heat distortion (ASTM D648) 350-400 F; impact strength 0.30-0.35 ft-lb Izod; compressive strength 18,000-21,000 psi; flexural strength 6500-10,000 psi; tensile strength 3000-4000 psi; Barcol hardness 58-65; resistant to weak acids; substantially no effect produced by hydrocarbon solvents and alcohols; attacked by alkalis

Application: Electronic components, ignition system parts, electric switches and instruments, magnetic motor starters.

For more data circle MD 26, Page 179

Overload Clutch 27

Flaton Machine Works, 7829 S. Broadway, St. Louis, Mo.

Designation: Type CL Safety Clutch

Style: Torque controlled overload release; automatic electric current shut-off or signal; shaft coupling style

Size: Bore ¾, 1⅛, 1¼, 1⅜, 1 13/16, 2⅝, 3⅝ and 4½ in. diam max; length 4¼, 5¼, 5 11/16, 7, 8⅛, 9⅛, 12¼ and 16 in. respectively; overall diam 2½, 3 7/16, 4, 4⅝, 6, 7⅛, 9⅝ and 12 in. respectively

Service: Overload or sudden shock releases load and either shuts off current or signals overload; resumes peak load when overload removed; operates at all speeds to less than 1 rpm; operates clockwise or counterclockwise; allowance for shaft misalignment

Design: Fiber friction cone, held in place by neoprene pressure ring, receives power from housing; pressure on cone, set by adjustment nut bearing on pressure ring, determines max load; on overload, friction cone ceases to rotate; housing continues to rotate and pin on housing rotates a shut-off spool on friction cone hub, causing spool to move axially along threaded section on hub, tripping limit switch; extended, demountable and integral hub designs available.

For more data circle MD 27, Page 179

Magnetic Starter 29

Euclid Electric & Mfg. Co., Madison, O.

Designation: Bulletin 5303

Style: A-c magnetic with 3-pole air circuit breaker and magnetic blowout; 3-wire

Size: Size 0, 1, 2—24 5/16 in. high x 9⅞ in. wide x 6⅝ in. deep; size 3—31¾ in. high x 15¾ in. wide x 10 in. deep; size 4—41 in. high x 17½ in. wide x 10 in. deep; all with 1 5/16 in. handle projection

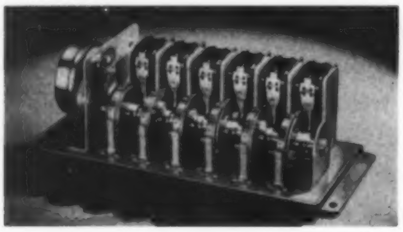
Service: Full voltage, nonreversing starting and overload protection of squirrel-cage type motors and primary switches of wound rotor induction motors; 25-60 cycles; magnetic blowout protection on all but size 0; air circuit breakers trip instantly on short circuit, after safe interval on overload; thermal overload relays provide inverse time overload protection

Design: Main contacts line type self-cleaning hard drawn copper, interchangeable, except size 0 has mill type contactor; double break silver-to-silver bridging type vertical auxiliary contacts mounted at right angles to shock; snap action bimetallic disk thermal overload relay, sizes 0, 1, 2; melting alloy type thermal overload relays, sizes 3, 4, 5; flexible braided copper shunts, size 0 transparent plastic insulated finger shunts; circuit breaker handle in front cover interlocked with door; self-lubricated graphite bronze bearings.

For more data circle MD 29, Page 179

Recycling Timer 28

Industrial Timer Corp., Newark, N. J.



Designation: Series MC

Style: Single synchronous motor drive; multiple circuit; adjustable; SPDT

Size: Bank of switches to suit

Service: Constant common time cycle from 1 rev in 40 sec to 1 rev in 72 hr; cams individually adjustable for *on* and *off* electrical periods from 2% to 98% of time cycle; timing sequence of cams independently adjustable; 15 amp micro type switches

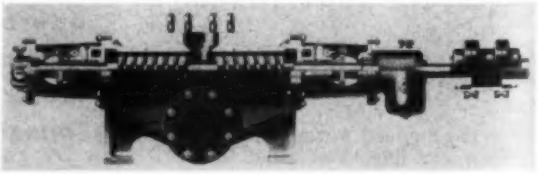
Design: Common cam shaft driven by synchronous motor; time cycle changed by replacing rack assembly in gear train between motor and shaft; drum calibrations from 0 to 100 indicate timing sequence; 0 to 100 calibrations on cam face indicate *on-off* periods

Application: Manufacturing and processing machinery.

For more data circle MD 28, Page 179

Screw Pump 30

Warren Steam Pump Co. Inc., Warren, Mass.



Designation: Warren-Quimby

Style: Opposed screw; externally geared; vertical or horizontal

Size: Up to 3000 gpm

Service: Positive displacement of non-lubricating liquids of low or high viscosity at temperatures to 800° F; pressures to 1000 psi.

Design: Combination stuffing box and bearing housing bolted and dowelled to main body casting; enclosed oil housing, bolted to brackets, seals in gear lubricant; anti-friction bearings; seven rings of packing plus sealing cage; shafts free to expand axially for high-temperature operation; furnished in any machinable metal, jacketed or unjacketed

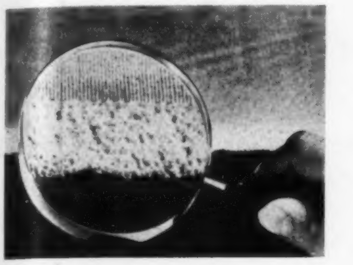
Application: Moving non-lubricating or viscous fluids such as water, brines, hot oils, chilled paraffin wax, distillates, sulphuric acids, sludges, molasses, acetates.

For more data circle MD 30, Page 179

NEW PARTS

Fabric—Foam Latex 31

Andrews-Alderfer Processing Co. Inc., 127-3 *Ash St., Akron 8, O.*



Designation: Andal

Form: Foam latex imbedded fabric, any desired color or shade

Size: Thicknesses from 1/16-in. to ¼-in. bonded to fabric on one side; continuously rolled lengths up to 75 yd, widths up to 60 in.

Service: "Breathing" qualities maintained since natural porosity not sealed; furnished in desired degree of density; withstands washing or boiling

Application: Instrument mountings or industrial filters in thin gages; sound deadening inserts.

For more data circle MD 31, Page 179

Electrical Steel 32

Armco Steel Corp., Middletown, O.

Designation: Tran-Cor T-O-S

Form: Iron-silicon alloy steel with high degree of orientation

Size: 4 mil thickness; 12⅜ in. wide coils

Service: For laminated core pieces operating at high cycles

Properties: 1800 minimum permeability at 18 kilogausses; density 7.65 grams per cu cm; volume resistivity 47 microhm per cm or 282 ohm per mil ft; lamination factor of 95% solid; chemically and thermally processed to reduce interlaminar energy losses; surface treatment unaffected by annealing; can be bent flat over small radius, then straightened without cracking

Application: 400-cycle wound type transformers and reactors on airborne electrical equipment.

For more data circle MD 32, Page 179

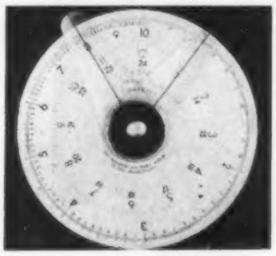
ENGINEERING DEPARTMENT

EQUIPMENT

For additional information on this new equipment, see Page 179

Time-Cost Computer 33

Willer Sales Co., P.O. Box 10008, International Airport, Los Angeles 45, Calif.



Designation: Loyd Computer

Style: Plastic circular disk showing time and log scales with two movable plastic index arms.

Size: 4 inches diam.

Service: Quick conversion from time to monetary values without intermediate computation.

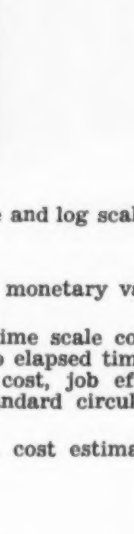
Design: Two concentric scales; inner time scale converts actual clock hours or minutes to elapsed time; outer log scale used for computing cost, job efficiency or relative proportions by standard circular slide rule method.

Application: Time study, motion study, cost estimating, production scheduling.

For more data circle MD 33, Page 179

Oscillograph 34

Precision Apparatus Co. Inc., 92-27 Horace Harding Blvd., Elmhurst, L. I., N. Y.



Designation: Series ES-500A

Style: 5 in. cathode ray laboratory type

Size: 8¼ x 14½ x 18 inches

Service: Vertical amplifier response beyond 1 mc, with 2 megohm input resistance and 20 mmfd capacity; horizontal amplifier response beyond 1 mc, with ½-megohm input resistance and 20 mmfd input capacity; vertical amplifier sensitivity better than 20 mv per inch with x1, x10 and x100 step-attenuator

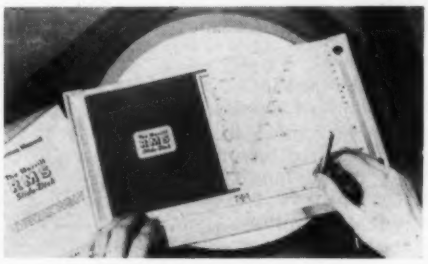
Design: Direct coverage from 10 cycles to 30 kc with internal linear sweep circuit; inversion of test patterns through phase reversing switch; internal phasable 60 cycle beam blanking plus input terminal for "Z" axis beam modulation; built-in horizontal phasing control; audio monitoring phone jacks provided

Application: Industrial electronics testing where high sensitivity is needed.

For more data circle MD 34, Page 179

Statistical Calculator 35

Graphic Calculator Co., 633 Plymouth Ct., Chicago 5, Ill.



Designation: RMS Slide-Disk

Style: Graphic triangular representation

Size: 10 in. disks; 12 in. wide x 8 in. high x 1 in. thick

Service: Performs operations of squaring, summing squares and square rooting for computation of standard deviations, root-mean-squares, correlation coefficients, least-square regression lines, skewed probability distributions; accuracy better than ½%; root-mean-square nomogram.

Design: Vinylite plastic; center of disk, which is free to rotate and slide vertically, indicates square root of sum of squares when successive values are marked on horizontal scale and brought to index position; furnished with 5 horizontal scales, nomogram straightedge, stylus, manual.

Application: Statistics, quality control, ballistics.

For more data circle MD 35, Page 179

High-Speed Recorder 36

Minneapolis-Honeywell Regulator Co., Brown Instruments Div., Wayne & Windrim Aves., Philadelphia 44, Pa.

Designation: ElectroniK

Style: Null balance servo system

Service: For measuring rapidly changing variables; records full scale signals up to 20 cycles per minute, signals with less than 10% of scale peak-to-peak amplitude up to 3 cycles per second; accuracy 0.045-mv for spans under 12 mv, 3/8% of span over 12 mv; dead zone 0.024-mv under 17 mv, 0.14% of span over 17 mv; chart speeds 1, 2, 3, 4 inches per second, 120, 240, 360, 480 inches per hour; for 115v, 60 cycle operation

Design: Emf from d-c source (thermocouple, tachometer-generator) is changed to a-c signal by a vibrating reed converter; circuit arrangement gives a-c signal output a definite phase relationship to line voltage phase, so that system can differentiate between increases and decreases in source voltage; a-c signal is amplified, and excites field winding of two-phase balancing motor, determining direction of motor rotation; motor drives pen and also moves contact on a measuring circuit slidewire to null position; adjustable damping circuit; motor-driven reroll for chart speeds over 1 in. per second

Application: Analyzing fuel efficiencies and engine designs in rocket engine testing, spectrographic analyses, determining instantaneous rates of flow.

For more data circle MD 36, Page 179

Contact KAYDON of Muskegon

FOR ALL TYPES OF BALL AND ROLLER BEARINGS: 4" BORE TO 120" OUTSIDE DIAMETER

KAYDON 4-Point Angular Contact Ball Bearings 22.350" x 29.125" x 1.687"

Raceways that make machines winners

Flame-hardening only the raceways of bearings, removes many handicaps in machine design. **KAYDON** pioneered this technique. Its success has helped make many machines winners!

KAYDON flame-hardening has made possible many special bearings of unusual design, like the 4-point angular contact ball bearings shown here ... with outer and inner races flanged for mounting directly to adjoining machine structures ... with gear teeth cut right on inner race, and inner races fortified by special protective finishes.

KAYDON flame-hardening of bearing-raceways to unusual depths (without through-hardening the races themselves) permits races to be drilled, tapped and gear-cut ... the bearing bulk and weight are greatly reduced ... many surrounding parts may be eliminated ... unusual shapes, very large diameters, extremely thin sections that facilitate mounting are achieved.

These are typical "winning" characteristics of **KAYDON** Special Bearings. Adaptable to many types of machinery ... maybe *yours!* Call **KAYDON**.

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HELPFUL LITERATURE

FOR DESIGN EXECUTIVES

. Control Valves

W. H. Nicholson & Co.—16-page illustrated talog No. 1250 deals with flat disk design ntrol valves in two, three and four-way pes for air, gas, oil, steam or water service. ver, solenoid and motor operated models available in wide range of sizes for pres- res up to 500 psi.

. Small Gasoline Engines

Cushman Motor Works, Inc.—Husky air- oled heavy duty gasoline engines in 1.5-3.2, 3-4.6 and 2.8-5.2 hp ratings are described 4-page illustrated bulletin E-3. Complete ecifications are given on units with manual sitive-action, automatic and disc type utches; transmissions with reverse gears; d other accessories.

. Flat Ceramic Capacitors

Sprague Electric Co.—Described in illus- ated bulletin 602, Bulplate flat ceramic pacitors are made in six sizes and in single d multiple capacitance combinations with ltage ratings up to 5000 v. Capacitors of e or more electronic circuits can be com- ned in single assembly to conserve space and ring.

. Hydraulic Pumps & Motors

Berry Motors, Inc.—4-page illustrated bro- ure 5010 and data sheets 1400CH-1 and 00CH-1 describe Berry principle of hydraulic wer as applied to pumps, motors and nsmissions, besides giving specifications of 00CH and 3200CH series units which func- n as pump or motor. Features of units are sented.

. Testing Facilities

Inland Testing Laboratories Inc.—24-page strated brochure covers test facilities, uipment and capabilities of this organization ich specializes in complete qualification ting of components and assemblies made der government contracts.

. Induction Motors

Ideal Electric & Mfg. Co.—Squirrel-cage in- ction motors are subject of 8-page illus- ated bulletin No. 200 which lists mechanical d electrical features of these units and ows these with cut-away picture. Dimen- ns are given for frame sizes ranging from 3 to 586.

. Starting Switch

Allen-Bradley Co.—8-page illustrated bulle- n 600 describes manual starting switch for actional horsepower motors. Typical in- allations are shown. Dimensional and en- neering data, features of each of six switch odels and variety of standard enclosures are cluded.

. Automatic Welding Process

Air Reduction—26-page illustrated reprint DR 66 is entitled "The Aircomatic Welding ocess." It discusses fundamentals of etal-arc transfer in inert-gas shielded-arc elding.

. Plastics

Polymer Corp. of Pennsylvania—Specifica- ns and sizes of nylon and Teflon in rod, rip and tubing forms are listed in illustrated lder. These materials have numerous ap- cations in electrical, metalworking and emical fields for machined or blanked rts.

. Flush Latches & Hinges

Hartwell Co.—24-page illustrated catalog de- ribes in detail the various styles of flush ches and hinges and includes data on ap- cation to tractors, instruments, air condi- ners, induction heating equipment and other oducts. In addition, plan drawings and mensions are provided for 16 types of latches d hinges.

. Lubrication

Fiske Brothers Refining Co.—56-page illus- ted data book 1-51 is guide to use of briplate specialized line of lubricants which intains wear-resisting film on surfaces of arings, gears, chains, cams, slides and other chine parts. Lubricant also protects parts ainst rust and corrosion while minimizing ction. Recommendations are given for uses all types of equipment and machines.

89. Magnetic Relays

North Electric Mfg. Co.—8-page illustrated temporary bulletin 1R511 "North Relays for Industrial Controls" lists relays according to their basic characteristics. Effort has been made to tabulate most widely used features. Specifications of types and table of available contact forms are included.

90. Acrylic Molding Powder

Rohm & Haas Co.—Recommended extrusion procedures to be used with improved Plexiglas VM acrylic plastic molding powder are presented in technical bulletin which also outlines advantages inherent in this material.

91. Self-Sealing Couplings

Aeroquip Corp.—42-page illustrated "Aeroquip Industrial Catalog" contains information on breakaway and self-sealing couplings, adapters, hose assemblies and fittings and accessories such as assembly tools, hose cover stripping tool and protective coil sleeves. How to order items and assembly instructions are included.

92. Aluminum Research

Aluminum Co. of America—16-page illustrated booklet tells aim and scope of research done at East St. Louis, Ill. research laboratory and nature of products Aluminum Ore Co. subsidiary provides for industry. These products include aluminas and fluorides.

93. Centralized Lubrication

Farval Corp.—"Studies in Centralized Lubrication" is 8-page illustrated bulletin which outlines production and operating economies attained by using centralized lubricating systems on open gearing, dredges, rolling mill, crusher and sugar mill. Case studies describe installation and show savings.

94. Electric Motors

Wagner Electric Corp.—4-page illustrated data sheet G51-1 presents application information and specifications of various types of single-phase, polyphase, squirrel-cage, direct current and gear motors.

95. Casting Holder

Joy Mfg. Co.—Details of Casting Grip floor machine which securely holds 2 to 20-in. castings for cleaning and snagging operations are presented in 2-page illustrated bulletin 76-M. Machine is used in conjunction with fast swing grinders.

96. Locknuts

Townsend Co.—4-page illustrated folder depicts special advantages of two types of locknuts for aircraft and industrial usage. Nylok nut has tough nylon plug insert in one of hex faces as locking element. Tufflok nut contains treated hexagonal fiber washer as locking medium. Both are cold forged. Description and specifications are listed according to sizes and finishes in table form.

FOR MORE INFORMATION

on developments in "New Parts" and "Engineering Department" sections—or if "Helpful Literature" is desired—circle corresponding numbers on either card below

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97. Synthetic Crystals

Linde Air Products Co.—4-page illustrated booklet "Linde Synthetic Crystals for Industry" lists properties, available forms and uses of synthetic sapphire, spinel, titania, calcium tungstate, cadmium tungstate and fine alumina polishing powders.

98. Package Testing Service

United States Testing Co.—8-page illustrated booklet is descriptive of testing program for reducing packaging and shipping losses. Procedures outlined include those for testing packaged products weighing from 100 to 1000 lb and method for testing packaged products weighing less than 100 lb.

99. Production Facilities

Telechron Inc.—20-page illustrated booklet presents facilities for carrying through small intricate electro-mechanical assemblies from original design or prototype to mass production as an aid to manufacturers of military equipment.

100. Miniature Foot Switch

Simonds Machine Co.—2-page bulletin describes Linemaster Treadlite miniature foot switch. Switch is applicable to control of sound and transmission equipment, relays, solenoids, magnetic equipment, business machines and similar units that require instantaneous and accurate control of relatively low amperage loads.

101. Diecasting Finishing

American Wheelabrator & Equipment Co.—"Wheelabrating, the Faster and Better Way to Finish Die Castings" is title of booklet No. 794 which illustrates and describes actual case histories. Three major uses of process are achieving slightly roughened surface that will provide bond for finish coatings, removing fine burrs and thin flash, and eliminating porosity in metal itself.

102. Rust Prevention

Rust-Oleum Corp.—Illustrated catalog deals with general industrial rust prevention and presents directions related to uses under various exposures and other conditions which produce rust. About 70 color chips are designated, and company's sealers for materials other than iron and steel are listed also.

103. JIC Pneumatic Standards

Miller Motor Co.—Recently adopted Joint Industry Conference pneumatic standards are reproduced in full, with sample circuit, glossary of terms and two pages of standard symbols. Comparison of how Miller air cylinders meet standards are included in this 12-page booklet.

104. Stainless Tubing & Pipe

Carpenter Steel Co., Alloy Tube Div.—16-page illustrated "Corrosion Notebook" contains data on corrosion resistance of various types of stainless tubing and pipe. Intergranular, galvanic, atmospheric and pit type corrosive conditions are described.

105. Friction Materials

Raybestos-Manhattan, Inc., Equipment Sales Div.—Brake linings, brake blocks and special shapes of friction materials are covered in 24-page illustrated engineering bulletin 400. Specifications are included on 50 types of brake linings and blocks and on 22 different clutch facing materials including woven, molded, semimetallic, metallic and patented Vee-Lok types.

106. Blind Fasteners

Cherry Rivet Co.—8-page illustrated condensed catalog C51 contains description and specifications of standard Cherry blind rivets and rivet guns. Many applications of vibration-resistant blind fasteners are listed in detail.

107. Heat Transfer Data

Kold-Hold Mfg. Co.—Data on use of Platecoils as medium of heat exchange is included in this technical manual which provides engineer with principal data required in solution of heat transfer problems. Process of selecting and applying industrial heat transfer equipment is simplified by supplying charts and formulas to aid in basic calculations of industrial heating and cooling applications.

108. Specification Card

American Chemical Paint Co.—No. 920 Specification Card presents information concerning specification chemicals for government and its contractors as well as general information regarding chemicals required for protective surface treatments of metals.

109. Neoprene Belting

Baldwin Belting, Inc.—2-page catalog se tion describes Baldwin Brown and Whi Supertex neoprene belting which can be us for many types of light conveying and tran mission service.

110. Transformers

Standard Transformer Co.—Detailed dra ings, accuracy limits, ratio and phase ang curves and other engineering data are include on line of current, potential and meterin transformers in 32-page illustrated bullet S-501-B. Design information is given to a in proper selection and application of tran formers.

111. Cellulose Esters

Tennessee Eastman Corp. — 60-page ill trated book contains information on uses cellulose esters, compatability of these este with solvents, resins and plasticizers, a procedures of analysis. It is primarily laboratory manual for those interested coatings, lacquers, synthetic fibers and mol ing compositions.

112. Fastening Specialties

South Chester Corp.—Designed to offer i stalled economies in metal-to-metal and metal-to-wood applications, complete line fastening specialties is described in 6-pag illustrated folder SCO 12. Items availab from stock include blind rivets, anchor nut screw fasteners, adjustable pawl fasteners a door retaining springs.

113. Gage Blocks

Webber Gage Co.—Manufacture and use precision gage blocks is subject of illustrat catalog. In addition to descriptions of stand ard, metric, carbide, heavy-duty and ang gage blocks, details of fixtures and accessori and information on care of gage blocks a included.

114. Control Motors

Transicoil Corp.—Illustrated 16 x 22-i chart includes easy-to-read tables on physic dimensions and electrical characteristics fo 25 different miniature control motors. Helpf application hints are given to show how t connect for plate-to-plate operation, spee torque curve and connections to obtain desire rotation.

115. Self-Tapping Screws

Parker-Kalon Corp. — 24-page illustrated pocket-size booklet No. 480 gives essentials of P-K self-tapping screw selection, applicatio information, recommended hole sizes and cor responding drill size numbers.

116. Laminated Plastics

Synthane Corp.—4-page illustrated folder is designed to help engineers, specifiers and buyers select proper grade of laminated plastic sheets, tubes and rods. Tabular data is correlated so that all systems of grade specifications as drawn up by industry and various government agencies can be compared at once.

117. Closed Cellular Rubber

Great American Industries, Inc., Rubatex Div.—Revised and enlarged catalog and panel of actual samples of Rubatex closed cellular rubber aid potential user in selecting proper type. Catalog includes chart of properties to meet revised ASTM standards No. D-1056-49T, and sample panel contains twelve 2 x 1½-in. specimens of rubber showing variations in density and color.

118. Potentiometers

Helipot Corp.—Illustrated data sheet describes eight unusual potentiometer installations which called for minimum space requirements and maximum adaptability to installation and operating limitations. They are typical of specialized designs developed and produced to meet customer specifications.

119. Induction Motors

Allis-Chalmers Mfg. Co.—6-page bulletin 05B7550 details construction features of large two-pole squirrel-cage induction motors for boiler feed pumps, oil pipeline pumps, centrifugal blowers, descaling pumps and other high speed drives. Cross sectional view shows construction of stator winding, bearings, ventilation, squirrel-cage winding, rotor, stator and bearing brackets.

120. Speed Changer

Metron Instrument Co.—Illustrated data sheet No. 4B describes type 4B miniature variable-ratio speed changer with lever adjustment. Device is for application to timers, recorders, controllers, computers, indicating mechanisms and other low power devices.

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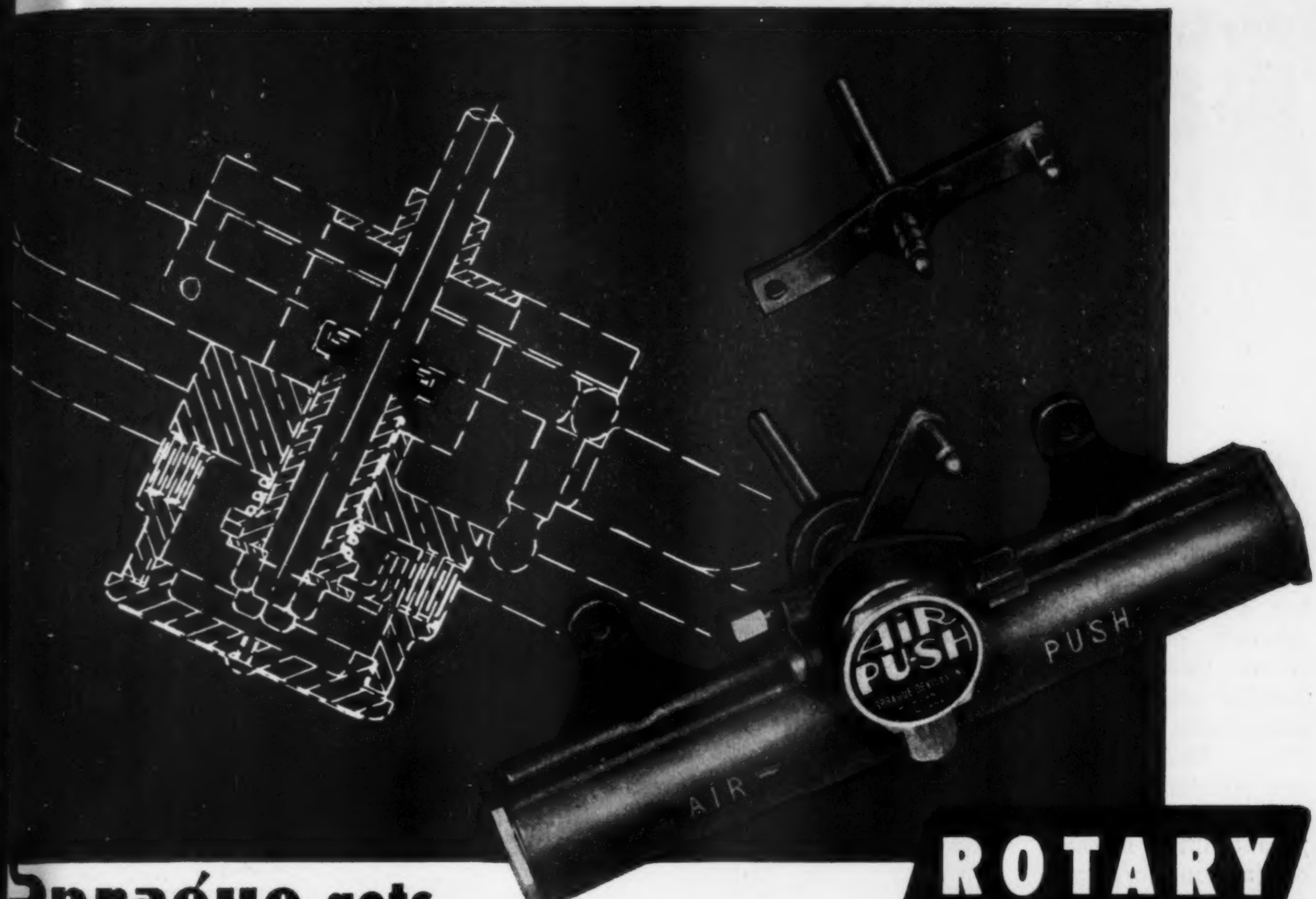

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MEN OF MACHINES

Formerly supervisor of design cost control for the Glenn L. Martin Co., **John A. Van Hamersveld** has joined Northrop Aircraft Inc., Hawthorne, Calif., as supervisor of producibility engineering. Mr. Van Hamersveld's new position will cover the functions of producibility liaison, castings and forgings, design producibility, design handbook information and producibility development projects. An employee of Glenn L. Martin for more than twelve years, he was previously associated with the Warner & Swasey Co. of Cleveland. Mr. Van Hamersveld has written numerous technical articles, three of which have appeared in MACHINE DESIGN within the last year.

John A. Van Hamersveld

♦

The James L. Entwistle Co. of Pawtucket, R. I., builder of machinery for the wire and cable industry, has announced the appointment of **Sidney E. Borgeson** as chief engineer. Mr. Borgeson's 28 years of experience in the design and development of wire and cable production machinery began when he became development engineer for Western Electric Co. While in this position he developed toroidal coil winders and other telephone equipment. Later, in the general research laboratories and the central engineering department of General Cable Corp., he designed several types of equipment, including high-speed magnet-wire flyers and special equipment for the Signal Corps and Navy. He has set up several plants and has developed equipment for the production of various types of wire and cable, the most recent of these projects being the setting up of an independent production unit for manufacturing 20,000,000 feet of wire monthly and planning the equipment for the production of spiral 4 cable. Mr. Borgeson is the author of many papers on developments and new methods in the wire and cable field and has acquired several patents. He is a member of ASME and of the New Jersey Society of Professional Engineers.

Sidney E. Borgeson

♦

At the 35th Annual Meeting of the American Gear Manufacturers Association **S. L. Crawshaw** was elected vice president. Mr. Crawshaw is presently assistant to the president of Western Gear Works and Pacific Gear and Tool Works, having served during the last eight years in the capacities of manager of the company's California division plant and manager of engineering and sales for all plants. Previously he was with Westinghouse Electric Corp., where he was manager of the gearing engineering department and chairman of the labor management committee. Mr. Crawshaw has developed and patented a rotary type counterbalanced crank for oil-field pumping units and a quill shaft arrangement for marine propulsion units. He is a member of the ASME, SAE and SNAME and has served as vice chairman of the Pacific Northwest chapter of ASME and as a member of the mechanical standards council of ASA. In addition to his new assignment as vice president of AGMA, Mr. Crawshaw is also chairman of the association's operating and control committee and for six years served as chairman of the general engineering committee. He has presented numerous papers and articles before AGMA and in trade jour-

S. L. Crawshaw

nals and prepared the Gearing Section of *Kent's Mechanical Engineers' Handbook.*

♦

John Seagren

John Seagren has been appointed chief engineer of the engine division of The National Supply Co. at Springfield, O. At the same time, the appointments of **Harvey W. Hanners** and **George F. Noltein** as chief research engineer and consulting engineer, respectively, were also announced. A native of Sweden, Mr. Seagren acquired extensive experience in the shops, testing departments and engineering offices of two Swedish concerns. After coming to the United States he joined Fairbanks-Morse and Co. as a designer, later becoming a development engineer. He became associated with the Atlas Imperial Diesel Engine Co. in 1931 and was placed in charge of diesel development and research. Five years later he was made chief engineer, assuming full charge of all experimental work and engineering. Mr. Seagren left Atlas to take over the duties of development engineer on post-war production at the Northern Pump Co. of Minneapolis and then joined the American Locomotive Co. In January, 1948 he rejoined Atlas, becoming a member of The National Supply Co. in July, 1950, when that company purchased the engine division assets of Atlas Imperial. Mr. Noltein has been with the company since 1927 and has served as chief engineer and director of research until receiving his recent appointment. Mr. Hanners became associated with National Supply in 1941 as a development and research engineer and has since occupied the positions of assistant to the chief engineer and chief engineer.

♦

Harvey F. Gerwig has joined Weston Hydraulics Ltd. as chief design engineer. Mr. Gerwig formerly was associated with Consolidated Vultee Aircraft Co. as pneumatics design engineer and has authored articles on high-pressure pneumatics for MACHINE DESIGN.

♦

Announcement has been made of two new appointments in the general machinery division of Allis-Chalmers Mfg. Co. **J. F. Roberts** has been named director of engineering, and **R. C. Allen,** consulting engineer. Mr. Roberts entered the Allis-Chalmers graduate training course in 1919 and was in the hydraulic department until 1927, when he left to become hydraulic engineer with the Power Corp. of Canada. In 1936 he was named head hydraulic engineer for the Tennessee Valley Authority at Knoxville. Six years later he returned to Allis-Chalmers as manager of the hydraulic department, the post which he held until his new appointment. Mr. Allen joined the company's steam turbine department in 1936, after serving in executive engineering capacities for a number of manufacturing concerns, including the Westinghouse Electric Corp., A. O. Smith Corp. and the Murray Iron Works Co. He served successively as chief engineer, assistant manager, and manager and chief engineer of the steam turbine department until 1947, when he became manager and chief engineer of the new turbopower development department, which has been engaged in turbopower research and development for mobile and stationary power plants.

♦

Formerly chief engineer and works manager, **J. J. Rozner** has been elected vice president in charge of operations of the Aetna Ball and Roller Bearing Co., Chicago. As manager of operations he fills a new post created by the company's rapid growth and product diversification program and will direct all manufacturing, engineering, inspection, quality control and development activities. Mr. Rozner joined Aetna's engineering department in September, 1928. In 1934 he was appointed metallurgist and a few months later was named assistant chief engineer. He became chief engineer in 1940 and took over the added duties of works manager in December, 1948. **J. E. Dillon** replaces Mr. Rozner as chief engineer, and the position of works manager has been dropped from the company's roster. Mr. Dillon joined the company ten years ago as assistant metallurgist and was advanced to metallurgist in 1945.

♦

The Timken Roller Bearing Co. has promoted **L. A. Holder** to the position of chief mechanical engineer and **C. M. Maratta** to chief consulting engineer.

♦

The Westinghouse Electric Corp. has announced the appointment of **Dr. S. W. Herwald** as engineering manager of the special products development division. Dr. Herwald joined the Westinghouse graduate student training course in 1939 and was assigned later to central engineering. In 1946 he became section manager in special products engineering, and earlier this year was made development engineering manager of the department.

♦

According to recent announcements from the General Electric Co., **Max I. Alimansky** has been appointed assistant manager of engineering for the transformer and allied product divisions at Pittsfield, Mass.; **Gordon E. Walter** has been named assistant division engineer of the specialty transformer engineering division at Fort Wayne, Ind.; and **Dr. Louis T. Rader** has been appointed assistant manager of engineering of the company's control divisions at Schenectady, N. Y. Mr. Alimansky was a co-operative student at General Electric while studying at MIT and became associated with the capacitor engineering division in 1929. Both Mr. Walter and Dr. Rader joined GE on the company's test course soon after graduation from college.


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Strength of Materials

By Max M. Frocht, research professor of mechanics and director of the experimental stress analysis program, Illinois Institute of Technology; published by the Ronald Press Co., New York; 420 pages, 6 by 9 inches, clothbound; available through MACHINE DESIGN, *$5.50 postpaid.*

An outgrowth of an earlier textbook by Riggs and Frocht, this new *Strength of Materials* is substantially different in approach and arrangement. Discussion of two basic types of failure—excessive deformation and fracture—introduce the subject. Their prevention is the objective kept in sight as stresses, strains and deflection are discussed for a wide range of conditions. Practical significance of stress and strain in design is emphasized, and frequent references are made to modern experimental analysis methods such as the use of strain gages, brittle lacquers, and photoelasticity.

Two early chapters discuss statically determinate stresses in tension and bending and those due to internal pressure and torsion. Then, basic concepts of strain and axial deformations and the states of stress and strain at a point are introduced. Next, the significance of static stresses and strains in design is discussed with the aid of experimental methods. The remaining half of the book treats such subjects as: shear and bending moments, stresses in symmetrical beams, failure under steady and alternating stresses, torsion in circular shafts, deflections of beams, statically indeterminate beams, columns and curved beams, and riveted joints and nonhomogeneous beams. Examples and problems are included with most chapters.

□ □ □

Electronic Motor and Welder Controls

By George M. Chute, application engineer, General Electric Co.; published by McGraw-Hill Book Co., New York, N. Y.; 348 pages, 6 by 9 inches, clothbound; available through MACHINE DESIGN, *$6.50 postpaid.*

Electronic controls for resistance welding and various types of electronic motor controls are covered in the two main sections of this book. Drawing heavily on his experience with General Electric, the author has given detailed explanations of both the theory and actual circuit operation of electronic controls of several manufacturers. In the description of more complex circuits, knowledge of simpler electronic con-

trols is presupposed, but adequate references to background material are included.

Resistance welding controls are considered in the first section. Basic welding controls, sequence welding and pulsation welding are dealt with in the first four chapters, with many specific examples given. The remaining seven chapters discuss synchronous timing, slope control, temper welding, forge timing, seam welding, welding with limited power supply and three-phase welding.

The second main section of the book, roughly half the book in length, considers motor controls for positioning, speed and register. These three basic motor control operations are covered in the first three chapters of this section. The rest of the section describes methods of accomplishing these basic controls and their use on various types of machinery.

Thus, three chapters discuss the G-E Thy-mo-trol, the motor-generator control of speed and the electronic amplidyne—all devices for controlling speed. Presented in the next two chapters are methods of maintaining speed stability and the photoelectric tracer, an electronic control for automatically following a pattern or template. The last three chapters describe electronic controls for specific types of machinery used in rubber calendering, multicolor printing and paper making.

Society Publications

Review of Current Research: Published by the Engineering College Research Council of the American Society for Engineering Education, this 1951 edition outlines policies and activities of engineering research in the 91 colleges and universities holding membership in ECRC. In addition to complete titles for the more than 5200 research projects now active, the volume shows for each school the names of responsible research administrative officers, a brief digest of policies which govern research projects and contracts at each institution, the number of personnel engaged in research activities, the annual expenditures, and special conferences and short courses of interest to research workers. A complete index of research project subjects, including over 4000 entries, facilitates use of the 250-page book. Copies of the publication, at $2.25 each, may be obtained from the Secretary of the Engineering College Research Council, Room 7-204, 77 Massachusetts Ave., Cambridge 39, Mass.

New Standards

Inspection of Fine-Pitch Gears ASA B6.11-1951: Tolerances and inspection methods for fine-pitch gears, defined as gears of 20 diametral pitch or finer, are specified in this standard. Developed originally as AGMA 236.02, the standard calls for inspection measurements under conditions closely approaching those of actual operation. The gear and a master, both mounted on a variable center distance fixture, are rotated in intimate contact. Displacements or variations in cen-

ter distance are measured or charted, giving a composite check of the combined effect of errors as explained in the standard. Two tolerance classification groups, commercial and precision, are arbitrarily designated. Each of these contains several classes, each based on the maximum permissible error. Allowable backlash value is given by a letter suffix which is applicable to any class.

Copies of this 35-page standard may be obtained from the American Society of Mechanical Engineers, 29 West 39th St., New York 18, N. Y., at $2.50.

Steel Butt-Welding Fittings ASA B16.9-1951: This 14-page standard covers overall dimensions, tolerances and marking for steel welding fittings. The present edition is a revision of the original 1940 standard. The former range of sizes has been extended to 24 inches. Reducing tees and heavy wall caps have been added to the standard. Copies may be obtained from the American Society of Mechanical Engineering, 29 West 39th St., New York 18, N. Y., at $0.75 per copy.

Applying Torque Converters

(*Concluded from Page* 144)

that of *Fig.* 18. One car transmission so constructed reports efficiencies in the coupling range as high as 95 to 96 per cent, not far from that obtained by a well designed coupling, and through the upper half of the converter range values of 85 to 90 per cent and more are reported.

It might be well to digress for a moment to discuss "staging" as distinguished from the device just discussed for which the term "phase" has been used. It is not usual to use both, and all constructions here discussed have one stage. By using two sets of turbines and stators the fluid can be caused to act twice on the output shaft, reducing the flow or curvature requirements for the same energy transfer. Note that this is not a splitting up of one wheel but the introduction of additional wheels. Fluid flow is from the impeller through the first turbine, first stator, second turbine and second stator, and back to the impeller. Both turbine wheels are secured to the same shaft, the vector-cycle is simply repeated. The action is similar to staging in a conventional hydraulic turbine.

Staging improves converter efficiency in conversion range, particularly at high ratios but at an appreciable sacrifice of first cost and of efficiency in coupling range. Inasmuch as the crane application indicates extensive coupling range use, staged converters are not indicated and will not be discussed.

If a coupling has a fixed stator it is said to be a single-phase unit. If the stator free wheels it has two phases. If the stator is split into two free wheeling units it is three phased; and if, in addition, the turbine is in two elements, giving five elements in all, the converter is said to have four phases. The unit employed in this application has three phases.

The next part of this series on converters will present actual converter construction, applying the idealized concepts to actual units, and will discuss in more detail the application described in Part 1.

Designing Noncircular Surfaces

(*Continued from Page* 114)

$$y_a' = y_a + R_a \sin \beta_a \qquad (25)$$

Similarly for the driven cam, from Equations 12 and 14 and *Fig.* 4*b*,

$$\tan \delta_b = \frac{r_b d\theta_b}{dr_b}\frac{\pi}{180} = \frac{\pi}{180}\left[C - \frac{C}{1 + \dfrac{K_a}{K_b} f'(P_b)}\right]$$

$$\times \left[\frac{K_b\left[1 + \dfrac{K_a}{K_b} f'(P_b)\right]^2}{C\dfrac{K_a}{K_b} f''(P_b)}\right] \qquad (26)$$

and simplifying

$$\tan \delta_b = \frac{\pi K_b^2 \dfrac{C}{r_a}\left(\dfrac{C}{r_a} - 1\right)}{180\, K_a f''\,(P_b)} \qquad (27)$$

From *Fig.* 4*b*,

$$\alpha_b = \theta_b - \delta_b \qquad (28)$$

$$\tan \alpha_b = \frac{\tan \theta_b - \tan \delta_b}{1 + \tan \theta_b \tan \delta_b} \qquad (29)$$

Substituting known factors,

$$\tan \alpha_b = \frac{\dfrac{y_b}{x_b} K_a f''(P_b) - \dfrac{\pi K_b^2 C}{180\, r_a}\left(\dfrac{C}{r_a} - 1\right)}{K_a f''(P_b) + \dfrac{y_b K_b^2 \pi C}{x_b 180\, r_a}\left(\dfrac{C}{r_a} - 1\right)} \qquad (30)$$

where

$$y_b = r_b \sin \theta_b \qquad (31)$$

$$x_b = r_b \cos \theta_b \qquad (32)$$

Finally,

$$\beta_b = \tan^{-1} - \left[\frac{K_a f''(P_b) + \dfrac{y_b \pi K_b^2 C}{x_b 180\, r_a}\left(\dfrac{C}{r_a} - 1\right)}{\dfrac{\pi K_b^2 C}{180\, r_a}\left(\dfrac{C}{r_a} - 1\right) - \dfrac{y_b}{x_b} K_a f''(P_b)}\right] \qquad (33)$$

Therefore, the location of the center of the cutter can be expressed as

$$x_b' = x_b - R_b \cos \beta_b \qquad (34)$$

$$y_b' = y_b - R_b \sin \beta_b \qquad (35)$$

For brevity, the numerical substitutions in Equations 23 to 25 and 33 to 35 have been omitted since they are straightforward and add little to the discussion. Equations 23 and 33 appear at first inspection to be cumbersome, but actually many of the terms repeat themselves. Organized in tabular form, the calculations can be completed rather quickly.

The satisfaction attainable in the final cams is determined by the accuracy of the computations and by

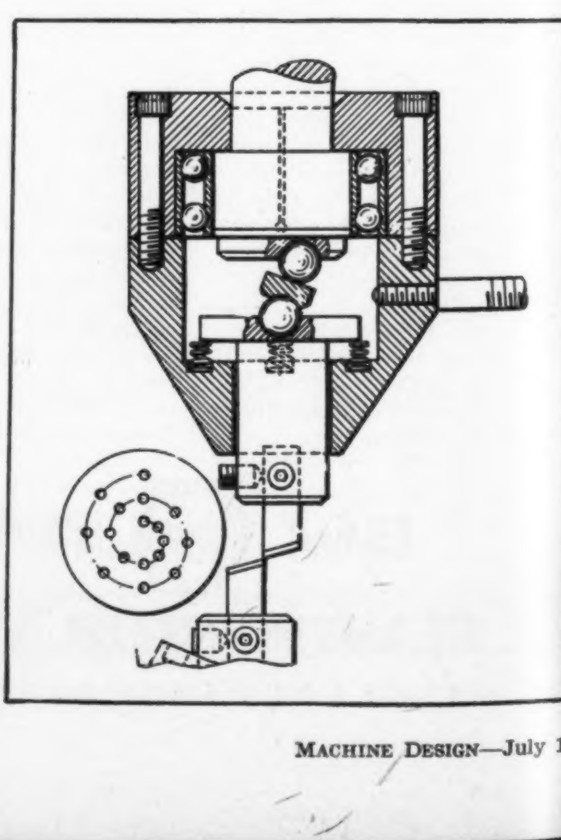
the precision of the machinery employed. Often a cardboard model is invaluable in determining the expected results and whether the geometry of the cams makes their operation and manufacture feasible. A cardboard model can be conveniently constructed by use of Equations 11, 12, 13 and 14 without concern for the more extensive computation of Cartesian coordinates.

REFERENCES

1. Peter Schwamb, Allyne L. Merrill and Walter H. James—*Elements of Mechanism,* John Wiley & Sons, New York, Fifth Edition, 1931, Section 8-13.
2. Francis Joseph Murray—*The Theory of Mathematical Machines,* King's Crown Press, New York, 1947, Section 2, Page 17.

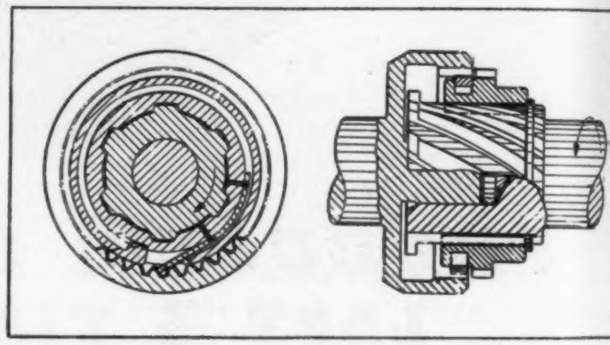
NOTEWORTHY Patents

CONVERSION of rotary motion to reciprocating motion in a direction parallel to and in the same axis as the rotary spindle is simply accomplished by the mechanism in patent 2,547,594, granted to Eric O. Ohlsson, Hagalund, Sweden. Lower end of the drive spindle or other rotating member carries an eccentric ball that contacts a plate supported by balls at two or more points on the reciprocating plunger. As the spindle rotates the plunger is reciprocated as the plate is rocked on its pivot points due to the eccentric location of the ball. The plunger is driven down as the driving ball rides over the center line between any two ball supports and is returned by springs when the driving ball approaches the an-



gular midpoint between any two supports. Number of plate supports determines the number of plunger strokes per revolution of the spindle; with two supports, the plunger reciprocates twice per spindle revolution. Plunger stroke can be changed by varying the amount of eccentricity of the driving ball in relation to the spindle axis. This device is particularly suited for converting drill press spindle rotation to a striking motion for plate shearing or hole punching.

SMOOTH TOOTH ENGAGEMENT of jaw teeth in an overrunning clutch is covered in patent 2,551,000. Internal splines on a sliding collar engage helical splines on the driveshaft, and external teeth on the collar can be engaged with teeth in a ring gear on the output shaft. When the driveshaft rotates faster than the driven shaft, a flat spring on the collar engages a tooth on the driven ring gear, preventing the collar from rotating with the driveshaft and causing



the collar to move along the helical splines and into engagement with the driven half of the clutch. If the driven shaft should start to overrun, a split ring, which engages the tips of the ring gear teeth, is rotated relative to the flat spring and forces the spring out of engagement with the ring gear. The collar is then free to slide along the helical splines and out of engagement with the teeth of the driven clutch members. Since teeth on the driving and driven elements are always synchronized when engaging, the clutch meshes smoothly, yet will transmit heavy loads even with small clutch members. The inventor, Winthrop S. Horton has assigned the patent to The Studebaker Corp.

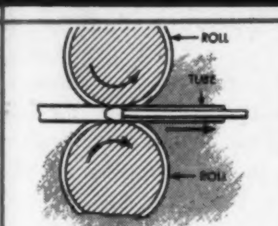
SELF-CONTAINED HYDRAULIC TAPPET for use on internal combustion engines does not require connection to the oil system of the engine. A light spring contained between the lower end of the tappet body and the bottom of a sliding plunger removes slack in the tappet system by urging the plunger up and admitting oil into the space below the plunger. When the cam forces the tappet up, a small disk valve in the lower end of the plunger is closed by oil pressure, trapping oil between the plunger and tappet body and opening the engine valve. A flexible rub-

let's take a CLOSER LOOK at

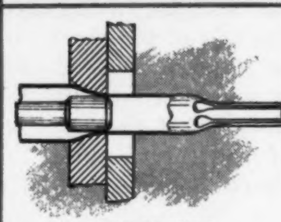
Surface Finishes

If you machine or fabricate hollow parts from tubing, chances are you have one or more *finishing* problems. Perhaps you can *start* finishing right in your own purchasing department by specifying the type of mill-finish best . . . and most economically . . . suited to your end-use requirements.

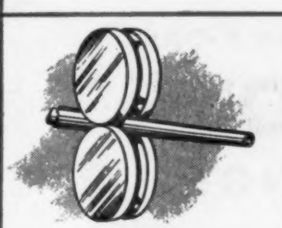
As a specialty tube mill, B&W can supply mechanical tubing with any of the following finishes — as an integral part of the manufacturing process:



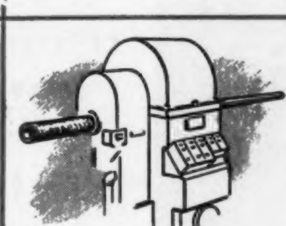
HOT-FINISHED
bears the scale formed during hot fabrication or heat treatment.



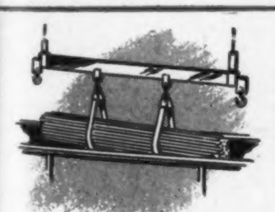
COLD-DRAWN
smooth, scale-free surface.



ROCKED
smooth surfaces, obtained by special sizing and finishing process.

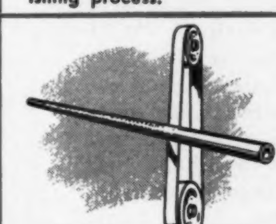


TURNED
machined, uniform O.D.



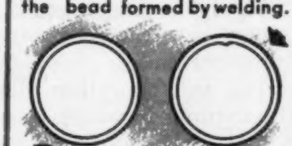
PICKLED
de-scaled by one of several solutions.

SHOT- OR SAND-BLASTED
O.D. and/or I.D. scale is removed by blasting.



POLISHED
O.D. and/or I.D. polished to one of several specified degrees of smoothness.

FLASH-IN
ERW tubing which retains the bead formed by welding.



FLASH REMOVED
ERW tubing, bead - free.
Both available either as welded or normalized.

Remember—tubing is not just bar-stock with a hole in it, but a semi-finished product, having a wide range of optional finishes, tolerances, chemical and mechanical properties. Ask Mr. Tubes — your B&W Tube Company representative — to help you select the tubing best suited to your particular applications.

B&W SEAMLESS and WELDED TUBES

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Executive Offices: Beaver Falls, Pa.
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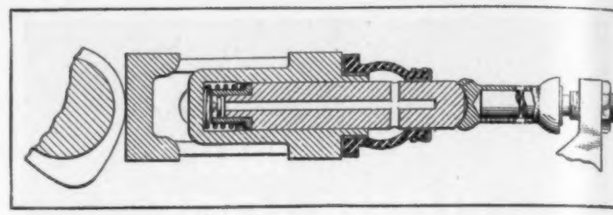
Alliance, Ohio—Welded Carbon Steel Tubing

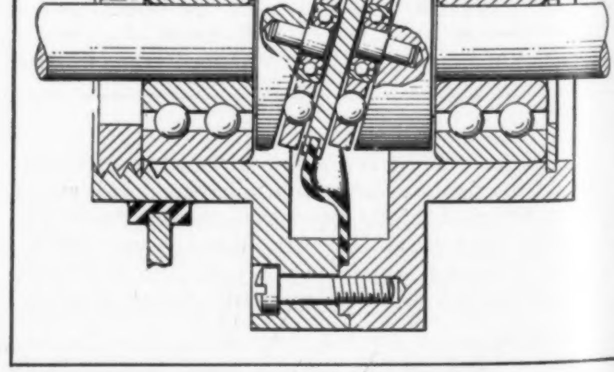
Sales Offices: Beaver Falls, Pa. • Boston 16, Mass. • Chicago 3, Ill. • Cleveland 14, Ohio • Denver 1, Colo. Detroit 26, Mich. • Houston 2, Texas • Los Angeles 15, Calif. • New York 16, N. Y. • Philadelphia 2, Pa. • St. Louis 1, Mo. • San Francisco 3, Calif. • Syracuse 2, N. Y. • Toronto, Ontario • Tulsa 3, Okla.

TA-1639M



ber sleeve between the upper end of the tappet body and the plunger contains the hydraulic fluid supply, making the assembly self-contained. If expansion of the engine valve-operating system should occur because of heating of the parts, excess oil between plunger and the bottom of the tappet body escapes back to the reservoir through the small clearance between the plunger and its bore in the tappet. The patent, No. 2,547,798, has been granted to Clyde W. Truxell Jr. and assigned to General Motors Corp.

MOISTURE-TIGHT COUPLING for preventing contaminants from entering electrical or other delicate apparatus through rotary shaft seals makes use of a flexibly mounted wobble plate barrier between input and output shafts. As shown in patent 2,545,562 assigned to Bell Telephone Laboratories Inc. by Felix A. Thiel, the coupling shafts end in enlarged hub sections having thrust faces in a plane inclined to the axis of the shafts. These inclined faces both bear on a thrust plate, through ball thrust bearings, which is vulcanized to the flexible rubber sealing barrier. Rotation of the input shaft causes a wobbling action of the thrust plate, because of wedging of the inclined faces, which drives the output shaft. Since



the flexible seal and thrust plate form an unbroken member across the inside of the coupling housing, moisture cannot get past the coupling into the interior of the motor housing or other casing.

Design of Integral Dashpot

(*Concluded from Page* 132)

lated operating time and the assumed operating time can be narrowed down by using smaller orifices and thus a greater number of them.

Test Results: Two integral dashpot hydraulic cylinders were made for tests—one with a tapered pin, the other with a straight hollow pin with radially drilled orifices. Numerous tests with the hydraulic fluid at room temperature and -45 F were made. The results of these undertakings with the tapered-pin design revealed an undesirable time-stroke characteristic curve but, also, the dashpot operating time increased many fold as the temperature of the fluid was lowered. The tapered-pin design under laboratory tests produced the following data: At room temperature, the dashpot operating time was 2.5 sec; under identical conditions, except with the fluid at -30 F, the dashpot operating time increased to 15 sec. The same laboratory tests showed that a radial-orifice dashpot of the design discussed here produced the following data: At room temperature, the dashpot operating time was 2.5 sec; under identical conditions, except with the fluid at -45 F, the dashpot operating time increased to 3.7 sec.

These experimental results are for an integral dashpot of a design with different dimensions from those shown in the sample calculation. Although the planned operating times are different for the dashpots of the sample calculation and of the test, design procedure was the same for both cases.

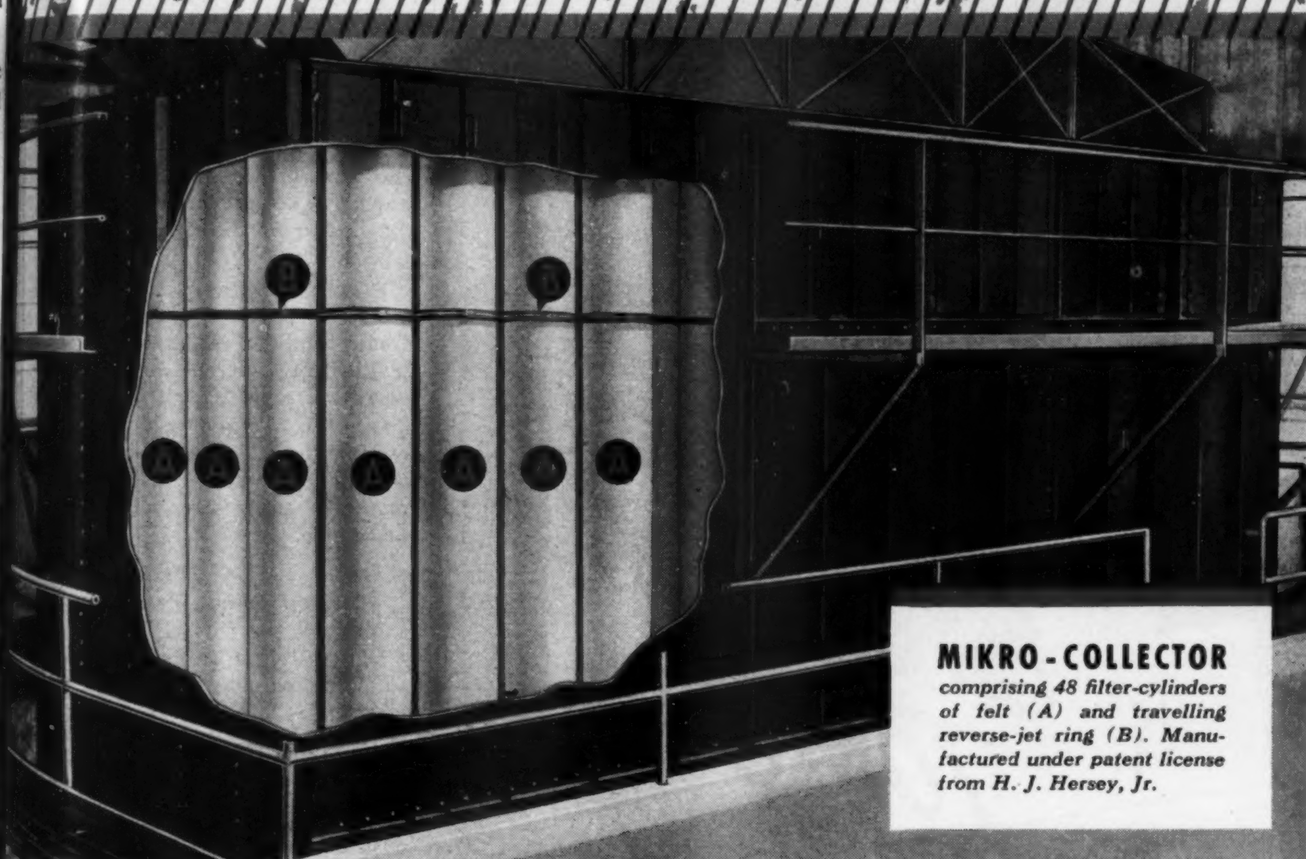
The severe temperature effect with the tapered-pin arrangement can be explained by the fact that the fluid motion in a thin annular orifice, as formed between a tapered pin and a recess port, assumes a shearing action. The more viscous the oil, the greater the required shearing force.

Conclusions: The relationship between the instantaneous orifice area to the dashpot stroke, as expressed in Equation 19, reveals that orifice area does not decrease linearly with decrease of stroke. Since the total time of operation should be a minimum, the amount of oil to be metered must be kept small. Consequently the total orifice area is small. This was verified by the sample calculation.

These results indicate that a similar design using a tapered pin is not as feasible because the small and nonlinear taper makes the pin difficult to fabricate. On the other hand, a straight pin with radially drilled orifices offers no severe manufacturing difficulties.

"Mechanical energy can be generated for about 1 cent per horsepower, but its equivalent in human energy costs $10 per horsepower. Higher living standards for the future depend on wider application of the efficiency of mechanical power."—William J. Kelly, *president, Machinery and Allied Products Institute.*

Optimum dust collection...



MIKRO-COLLECTOR
comprising 48 filter-cylinders of felt (A) and travelling reverse-jet ring (B). Manufactured under patent license from H. J. Hersey, Jr.

with AMERICAN Felt

The unique principles employed in the MIKRO-COLLECTOR enable dust recovery (up to 99.999%) and phenomenal filter rates. Pulverizing Machinery Company, Summit, New Jersey, the manufacturer, states that American Felt's wool felt has been found to be a superior filter medium, permitting the easy handling of damp or dry, light or heavy dust-loaded air streams. With the Hersey travelling reverse-jet principle uniform filter resistance is maintained, thus assuring uniform air flow. Used in the handling and recovery of a wide range of dusts and powders, the MIKRO-COLLECTOR serves to eliminate atmospheric pollution, as well as providing full recovery of a valuable product. Dangerous or noxious dusts and minute size dust particles are easily handled. Installations for the handling of ultra-fine radioactive dusts during the past year have a combined capacity of more than 100,000 cfm. The MIKRO-COLLECTOR is manufactured with single and multiple filter cylinders of varying diameters and lengths to meet every requirement.

American Felt Company

TRADE MARK

GENERAL OFFICES: 22 GLENVILLE ROAD, GLENVILLE, CONN. — ENGINEERING AND RESEARCH LABORATORIES: Glenville, Conn. — PLANTS: Glenville, Conn.; Franklin, Mass., Newburgh, N. Y.; Detroit, Mich.; Westerly, R I — SALES OFFICES: New York, Boston, Chicago, Detroit, Cleveland, Rochester, Philadelphia, St. Louis, Atlanta, Dallas, San Francisco, Los Angeles, Portland, Seattle, Montreal

Design Abstracts

(*Continued from Page* 162)

be ordered specially for part-winding starting.

The two parts of the winding usually are in a 1/1 ratio, but other ratios can be used, depending on how the particular motor coils can be connected. Many other schemes can be used, such as single-phasing steps, resistors, and others, which increase the cost. Closed transition is inherent. The cost index is 105, including motor modifications where required.

ELECTRONIC STARTER: The electronic starter in *Fig.* 1e consists of electronic tubes or contactors, on which phase-shift control can be used to delay the tube firing time. This reduces the average voltage, current, and torque. The starting torque is readily adjustable, but at a high price—if tubes are not otherwise justifiable.

Highest Torque Per Ampere

WOUND-ROTOR STARTER: The wound-rotor motor and starter illustrated in *Fig.* 1*f* are both basically more expensive than squirrel cage. But because the maximum obtainable starting torque per ampere is from this motor, it is included in the discussion.

The least expensive arrangement is a linestarter for the primary and a manual drum controller and resistor for the secondary. The cost index is 1.93. For pushbutton starting, a magnetic type starter is available with a primary contactor, the required number of secondary contactors, and a resistor. The cost index is 220. In addition to producing the highest torque per ampere, the drum controller or a modification of the magnetic starter can be used for a limited speed control.

COMPARATIVE CHARACTERISTICS: *Fig.* 2 shows the speed-torque characteristic of a squirrel-cage motor at rated voltage, together with three reduced voltage curves such as would be produced by the 80, 65, and 50 per cent voltage taps of an autotransformer. The dotted curve is the torque with a primary resistor or reactor starter, producing 80 per cent voltage at starting. It is apparent that the percentage torque increases as the motor accelerates. The star-delta and part-winding curves fall essentially parallel to those of the autotransformer starter, but at 33 and 50 per cent

torques. Showing torques available with various values of secondary resistance, *Fig.* 3 is a typical wound-rotor motor curve.

CONCLUSIONS: Of the types discussed, the manual autotransformer has the lowest initial cost. The part-winding starter is the lowest cost magnetic starter, and particularly useful where starting torques of the load are under 50 per cent of the full winding torque—or where increment starting is permissible.

Where starting torques must be higher, and where currents are allowed to be higher, the primary resistor or the autotransformer are needed. The primary resistor starter is more suitable to loads starting at low values and building up as the speed increases.

For the best in torque per ampere of starting current, the wound-rotor still is at the top, but it has also the highest cost. Therefore, it is seldom used, except where the last bit of torque is needed—or where speed control also is needed.

From a paper entitled "Cushion starting of A-C Motors," presented at the 15th Annual Machine Tool Electrification Forum sponsored by Westinghouse Electric Corp., in Pittsburgh, April 10-11, 1951.

Designing a Special Machine

By Frank J. Fink
President
F. J. Fink & Co.
Cleveland, O.

MACHINES fall loosely into three groups: standard machines which form the bulk of all machines and can generally be bought from a catalog; semistandard machines which include standard drilling, tapping, and a variety of other package components; and the special machine. The third category can be divided into two subgroups: the single purpose machine made in moderate quantity and peculiar to a particular industry; and special machines which are seldom built in a quantity of more than one. The latter present a unique problem to the designer, because the first machine built must be right. This group is the subject of our discussion.

In our proposal to a customer, we set forth some feasible method for doing our job, set up tentative specifications, and, probably, furnish perspective sketches showing the gen-

HOW YOU BENEFIT FROM THE WAY G-E SILICONE RUBBER RESISTS HEAT AND COLD

Important advantages to design engineers are offered by General Electric's new, improved silicone rubber, G-E 12650. Specifically formulated for applications requiring a high-heat-resistant material with greater elongation and hot tear strength than former silicone stocks, G-E 12650 provides a combination of heat resistance and mechanical strength far superior to previous grades. Serviceable between –85 and 500 F, the new G-E 12650 makes possible entirely *new* applications for silicone rubber.

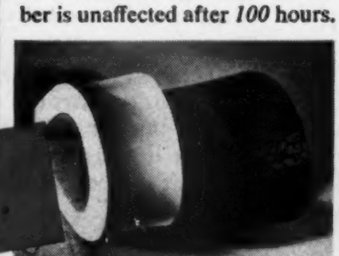
Heat-testing General Electric's new silicone rubber, G-E 12650, which resists temperatures as high as 500 F.

DEPEND ON G. E.

General Electric's *specialized* fabricating service is ideally equipped to supply the silicone rubber parts you need. This specialized service fabricates *only* silicone rubber compounds—your assurance of high-quality parts uncontaminated with other materials. Special molding and extruding techniques are employed, using either G.E.'s own quality-controlled compounds, or any commercially available compound specified by a customer. General Electric, a pioneer in the development of silicone rubber, has tremendous "know-how" gained through designing hundreds of applications. Put this *specialized* fabricating service to work for *you!*

Address inquiries to Section F-4, Chemical Division, General Electric Company, Pittsfield, Mass.

After exposure for 10 hours at 400 F, the surface of conventional rubber decomposes, while the surface of G-E silicone rubber is unaffected after *100* hours.



The sealing gasket for this oven door won't soften or lose resilience despite the high heat of the range. G-E silicone rubber was specified for this application.

DEPEND ON G. E.'S SPECIALIZED FABRICATING SERVICE

GENERAL ELECTRIC

eral functioning and expected appearance of the machine.

The designer must then transform these generalizations of word and picture into solid iron and steel. This is where the hard work begins. The way to design a machine is to make a start and do a little exploratory work on paper. We are at first little concerned with matters of strength and size, kinetic and harmonic precision. The experienced designer depends in these initial efforts largely upon his experience and skill in establishing proportions and devising movements. One of his most valued tools at this time is the humble eraser. Consultations are frequent. Changes in specifications are often required. No effort is made to make the first layout a good one. On the contrary, it is emphasized that the first drawing is for the purpose of investigation only and will not be used. In this way, the designer is freed from inhibitions and experiences a mental relaxation and flexibility that permits his thinking free rein. Under these conditions, that first layout often does turn out to be the final one.

When To Stop

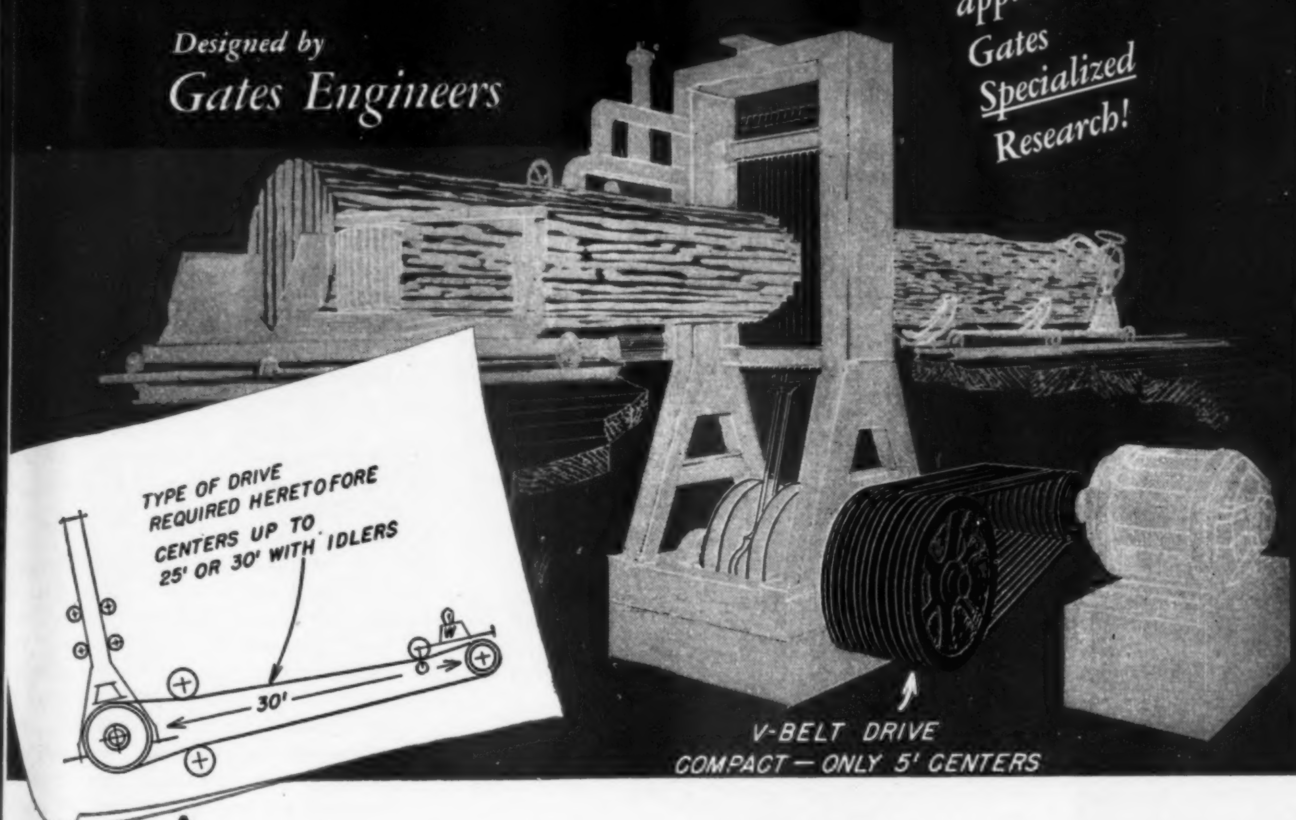
But the time must come when we have to narrow down, put on our blinkers, and stick to the narrow road. We take the risk, of course, that we might shut out an idea of merit but we cannot afford the many little side trips that on the average produce little improvement in return for the loss of time they cause. We have to bear in mind that we are designing a single machine. It must be right the first time, of course, but we cannot afford to strive for perfection; for if we do, it may be a long time before that machine is built. We make a distinction between right and perfect. Common horse sense has to be the keynote. If we were designing a machine for quantity production, our procedure would be different. For then we would indulge in functional models, at least one pilot model, and many tests before making the final drawings. If we were to use that kind of procedure for our "one only" machine, it would be easy to triple or quadruple the cost.

As the original plot develops on paper and the design begins to look good to the practiced eye, careful checks are made as to strength, physical limitations, and other theoretical requirements. At this point time is given to the mathematician and stress analyst. The designer has made preliminary calculations, but in the main his experience and instinc-

The first successful V-Belt Drive for GANGSAWS

Designed by
Gates Engineers

Another practical application of Gates Specialized Research!



A large maker of gangsaws said to Gates Engineers, "If you can design a V-Belt drive to handle the *fluctuating load* on a gangsaw, we can discard the long and troublesome flat belt that we have always thought we had to use."

The *reasons* for his statement are quickly told:—

A gangsaw (or Swedish type Gang Mill) consists essentially of several saw blades mounted in a heavy steel frame or "sash" that moves rapidly up and down like a monster jig saw. This sash may weigh a ton or more—yet it makes about *five* 24-inch *up* and *down strokes per second!*

On the *up* strokes, the saws have no cutting load on them. But on the *down* strokes they must often slash a log up to 36" diameter into 2" planks. This great unbalance in load between the *up* and the *down* strokes has heretofore defeated all efforts to apply a V-Belt Drive to saws of this type.

After thoroughly analysing the fluctuating load on gangsaws, Gates Engineers recommended very minor changes in design which would permit the application of V-Belts and—what is of *utmost importance to you*—they were able to write down *in advance* the exact specifications of the V-Belt Drive that has very successfully handled this extremely difficult assignment.

This *pre-solution* of drive design problems is possible because Gates operates the largest V-Belt testing Laboratories in the World—and Laboratory findings are carefully checked by tests made under actual field conditions. Finally, the *results* of these exhaustive tests are immediately reduced to *usable data* for the design of V-Belt drives to *perform whatever task may be required.*

Phone a Gates Field Engineer

If you have a difficult drive to design, or if some drive in your plant is giving trouble, or if you only want to be sure what size and construction of V-Belts will give the most efficient and the lowest cost service on any particular drive—you have only to phone a Gates Field Engineer, *always near you* in all industrial centers.

Just look in your phone book under "Gates Rubber." A Gates Field Engineer will come right to your plant and put at your service the full benefits of Gates V-Belt knowledge and experience *without the slightest obligation!*

ENG. 515

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IN ALL INDUSTRIAL CENTERS

THE GATES RUBBER COMPANY
DENVER, U. S. A.

tive proportioning play the importan role. As with other skills, trainin and practice have developed intuitiv directness and special talents. It i important, especially in times lik these, to question the availability o the proposed materials before the are frozen into the design. If i short supply, the materials may b ordered immediately.

We must now make such decision as shall it be mechanically, hydrau lically, pneumatically, or electrically actuated. We cannot afford the lux ury of fetishes in such matters. W have no place for the fellow who i a "this'er" or a "that'er". Each o the various means of accomplishing our design or transmitting power o motion must be selected on the basi of a particular fitness for our special requirement. Decision has to be based on what is going to do our job best at lowest cost. Mechanical movements are usually most satisfac tory for special machines; for them auxiliary components, such as pumps, compressors, and valves are reduced to a minimum.

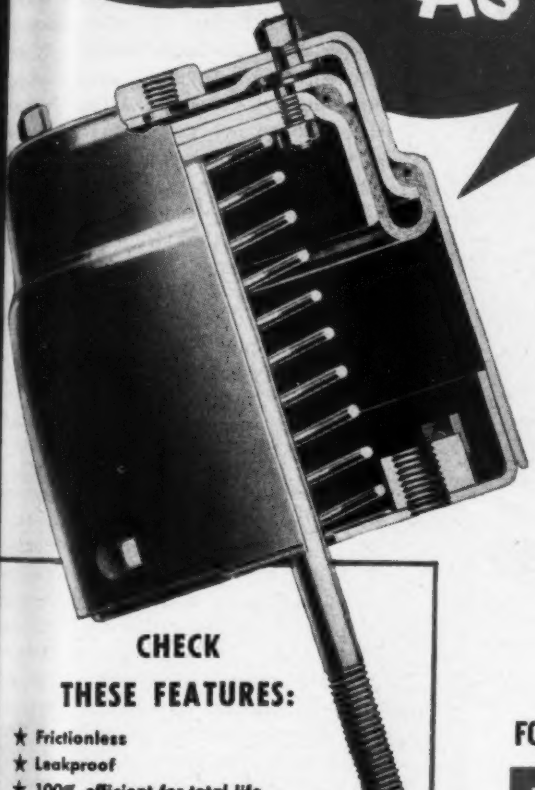
Go slow on new and unproved com mercial units, but take advantage of the benefits offered by the ones that have established their place. The opportunities for utilizing standard units is, of course, very limited in the special, special machine. But whenever they can be, not only are savings accomplished, but replace ment in an emergency will be great ly facilitated.

Unitize at All Stages

No matter how small a machine may be, it should be unitized. That is, the various functional groups should form as nearly an independ ent assembly as possible. On the larger machines this is a must. Most designers agree on these points over their coffee, but it is not always to be seen in practice. You have all seen a mechanic take apart a whole machine to get at a small shaft or pin or key or what's probably worse, to hear an indispensable production unit thumping and growling for weeks or even months with some chronic disorder because repair is so expen sive that it cannot be undertaken. Unitization is important first of all during the design of the machine. The various units can be assigned to different men, and the work speed ed up. This advantage follows through during manufacture and as sembly. These independent units may also be tested individually before as sembly. Unitization simplifies servic ing, repair and replacement. Unitiza tion is a great boon in the event part of the machine must be redesigned

ON OLD OR NEW MACHINE TOOL OPERATIONS—

YOU CAN STEP UP YOUR PRODUCTION AS MUCH AS 30%



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The pressure for production is on! Manufacturers everywhere find it increasingly difficult to keep pace with the demands of the day. But there is a way—a brand new way requiring no addition to manpower, no addition to plant facilities. The answer? Remarkable Bendix-Westinghouse Robotair Industrial Controls. With their wide range of applications, Robotair controls offer you amazing improvements in production speed and economy through increased productivity of man and machine at *unbelievably low initial cost.* Robotair units are available as original equipment for machinery manufacturers or for easy installation on shop equipment.

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- ★ Leakproof
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- ★ Diaphragm has rolling action and exceptionally long life
- ★ Made from high grade steel stamping—all metal parts cadmium plated
- ★ Chrome Molybdenum Steel Push Rod heat-treated for maximum toughness
- ★ Easily installed

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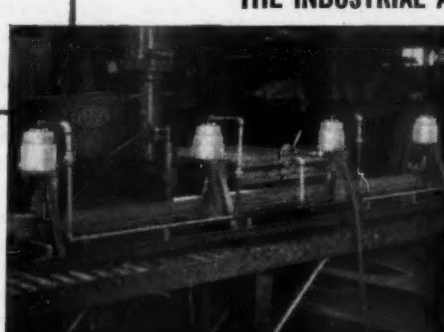
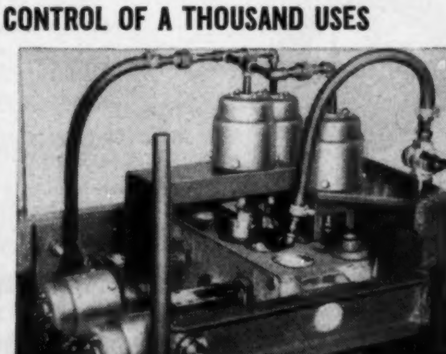


REDUCES OPERATOR FATIGUE

Four Rotochambers installed on the fixture in this spot welding operation replaced cumbersome hand clamps, saving time, greatly reducing operator fatigue.



SPEEDS INSPECTION METHODS

Installation of Rotochambers on this water test operation for engine castings eliminated hand operated fixtures, thereby speeding inspection and cutting costs.

either to improve its operation or because of changing conditions. Successful units of this type frequently become the standardized packaged units of industry. Unitization, furthermore, permits keeping on hand complete working units for quick replacement in an emergency.

What about the appearance of a machine? Should it be beautiful? Should it have eye-appeal? Or should appearance be completely ignored? Some might even prefer a machine, in particular a special machine, to be exactly the reverse of beauty. But this matter of beauty is a subject that in thousands of years has not been completely answered. Nevertheless, some things have almost universal attractiveness. It seems to me the best way to describe this phenomenon is that it has an appearance of rightness. I believe this is true also of the machine. You will note that such natural and well-ordered curves as the sine, parabola, catenary are beautiful. That is because they follow a law that is right. If a piece of mechanism is correctly designed, it will invariably have that right appearance.

Design for Safety

We cannot pass from design considerations without mentioning safety. It is sometimes disproportionately costly to design safety into a machine. Guards sometimes seem to add unnecessary costs to the machine. Why can't the operator value his own safety enough so that this expense will not have to be accepted. The answer to that question I don't know, but I do know that one of the greatest battles we have to face in the matter of safety is with the man whose life and limb we are trying to protect. It seems to be one of the perverse reflexes of human nature to have contempt for one's own safety. This may be a false notion of courage, but it has often led to tragic accidents. It is nearly always the fault of the operator, very rarely the fault of the machine when an operator is injured. But like with insurance, you have to face the cold facts and put up barriers against the man who persists in being hurt. Accidents are so costly both to the owner of the machine and to the operator that the expense of building safety into a machine is not only justified but is absolutely necessary, even if there were no laws to prescribe it. In spite of the greatest precautions the designer may take to provide fool-proofness in a machine, there will still be accidents. But the battle must go on.

From a paper, "Why a Special

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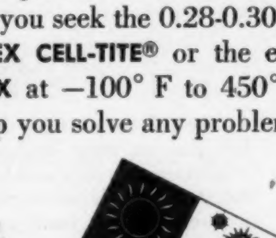
This new booklet on the properties of, test data on, and specifications for cellular rubber has just been released. It's concise, and a valuable reference source. Write for a free copy today.



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Machine and How," given at the Machine Design Conference of the Cleveland Engineering Society, Cleveland, O., February 5, 1951.

Properties of Materials under Extreme Pressures

By Percy W. Bridgman
Harvard University
Cambridge, Mass.

HIGH stresses with which we shall be concerned were, in the first instance, encountered as an incident to the generation of high hydrostatic pressures, and were, at first, merely incidental to an investigation of the effect of high pressures on the physical properties of different sorts of material. Formerly, the magnitude of the hydrostatic pressure that could be used in the laboratory was limited by leakage of the liquid with which pressure was transmitted. This limit, in the experiments of Amagat, was about 45,000 psi, which is approximately the pressure employed in heavy artillery, and is already above the working stresses permissible in many industrial applications. The work which I have to report to you was made possible by the discovery of a method of packing the joints of a pressure apparatus in such a way that the packing is automatically made tighter by the action of the pressure itself. I have called the principle of the packing the principle of the "unsupported area". The result is that the liquid cannot leak, no matter how high the pressure, which is now limited only by the strength of the containing vessel. This packing made it possible to raise the pressure at once from 45,000 to 180,000 psi for routine experimenting and, on special occasions, to 300,000 psi. This is beyond the range ordinarily reached in industrial processes and even beyond the stresses reached in tensile strength determinations of many ordinary materials.

In this new range of stresses, the steel of which the pressure vessels were constructed did not behave in the expected ways. In particular, the bursting pressures were higher than expected, and when the burst did occur, it started on the outside surface instead of the inside. Also, a paradoxical new kind of fracture was found in which a tensile type fracture took place across surface on which there was no stress. The occurrence of these unexpected features made it necessary to take time out

for study of the behavior of the steel vessels for their own sake, before the more academic program of investigating the general effect of pressure on physical properties could be resumed.

One result of some practical interest came out of this preliminary work. It was found advantageous to season the high pressure vessels by a preliminary application of pressure considerably in excess of the future working pressure. Under the high seasoning pressure, the inner layers of the vessel are strained plastically and given a permanent set, so that when pressure is released the external layers shrink back on the inner layers, leaving the internal layers in compression and the external layers in tension. The initial effect of any subsequent application of pressure is now to relieve the compressive stress at the internal layers, so that they are not put into tension until pressure is considerably raised. The net result is that the vessel responds elastically over a much wider range of pressure than on the initial application. The distribution of stress produced in this way is practically the same as that produced in a built-up gun by shrinking on hoops, but with the advantage that the stress distribution is smooth. By prestraining a pressure vessel in this way it is possible nearly to double the working range of pressure, and all my early work was done with vessels treated in this way.

Adapted to Artillery

It was in this connection that one of the few industrial applications of this academic work was made. It was an old idea to cold work the inside of heavy artillery by an initial stretching by a pressure much in excess of the firing pressure. The idea had never been reduced to practice because of the technical difficulties in applying sufficiently high pressures, particularly to a vessel in the process of stretching. However, by the use of the packing principle of the "unsupported area" the technical difficulties are easily surmounted.

The method has by now become, I am told, one of considerable importance in the production of certain types of artillery. The pressures used in gun stretching range up to 150,000 psi, which is, I imagine, the highest hydrostatic pressure used today in industrial application.

The upper limit of the pressures that can be attained in heavy steel vessels pretreated by internal stretching is, with present grades of steel, about 450,000 psi, a pressure which has been reached by D. M. Dewitt in London. However, it is not economical to try for such extreme pres-

sures because of the too short life of the apparatus. In my own work, I have found the useful working limit to be much lower—less, even, than 300,000 psi. In order to attain higher pressures, it is necessary to apply external support to the vessel to counteract partially the effect of the internal pressure. There are various ways of giving the vessel external support. Perhaps the simplest is to give the vessel a conical form and to push the whole vessel into an externally supporting conical sleeve simultaneously with the development of internal pressure. A particularly simple way of doing this is to use the same force which drives the piston, which generates internal pressure, to push the whole vessel at the same time into the supporting sleeve. The limit of pressures attainable with the device of this sort is, with present grades of steel, about 750,000 psi. The limit is set by the extrusion of the whole pressure vessel through the supporting conical sleeve.

External Support Required

In order to reach pressures materially higher than 750,000 psi the vessel must be more effectively supported. This means supporting it over the entire external surface, instead of only over the side walls. Such complete support is most simply provided by immersing the pressure vessel in a fluid carrying a high hydrostatic pressure. This amounts, essentially, to constructing one pressure vessel inside another. Theoretically it should be possible to reach any pressure by constructing a nest of pressure vessels, one within another, the pressure rising progressively from the outer to the inner vessels. There are obvious difficulties in the practical realization of such a scheme connected with the manipulation of the inner vessels through the outer vessels. Up to the present, only the first stage of such a scheme has been reduced to practice. By using approximately 450,000 psi as the external supporting pressure, and this means using an outer vessel with conical support, it is possible to reach 1,500,000 psi or even more in the internal pressure vessel.

In apparatus of this sort, it is possible to measure volume changes with acceptable precision, and to study the compressibility of materials and such polymorphic transitions as they may undergo. The volume changes produced by pressures of 1,500,000 psi may be quite large. The most compressible metal is caesium, which is reduced by this pressure to one-third its initial volume. The least

compressible substance, on the other hand, is diamond, which is reduced in volume by only 1.5 per cent by the same pressure.

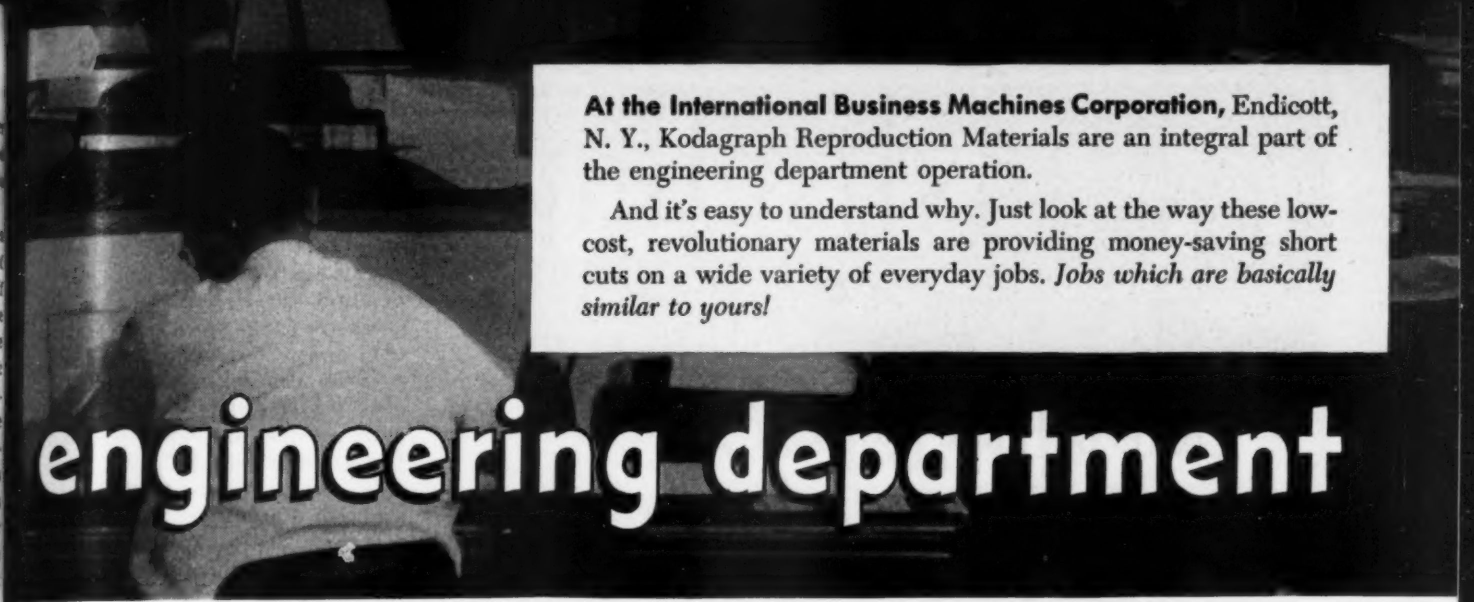
The vessels within which pressures of 1,500,000 psi are generated are of Carboloy instead of steel. Even if steel were strong enough, its use would be attended by the very grave disadvantage that the purely elastic distortion under such a pressure increases the internal diameter of the pressure vessel by 10 per cent. Carboloy, on the other hand, deforms elastically only one-third as much as steel, and a stretch of the vessel by three per cent is not prohibitive. In spite of this advantage, it would not be possible to use Carboloy if it were not for a most fortunate change in its properties produced by the pressure with which the vessel is supported. Under normal conditions Carboloy is almost as brittle as glass. It is nearly twice as strong in simple compression as the best steel, but in tension it is markedly less strong. When immersed in a liquid carrying pressures of the order of 400,000 psi, however, it loses its brittleness, and becomes threefold stronger in tension. This means that, when supported by external pressure, a Carboloy pressure vessel will support a tension at the inner layers which would normally fracture it, and thus can withstand much higher internal pressure than would otherwise be possible.

Pressure Increases Ductility

Steel, when subjected to hydrostatic pressure, undergoes a change in its physical properties similar to those experienced by Carboloy. It is found that a spectacular increase of ductility is produced by pressure. Under supporting pressures of 400,000 psi, steels which normally fracture in tension with a reduction of area of the order of 50 per cent, will permit practically indefinitely large reductions, so that the neck can be drawn to a sharp point before fracture, as can some substances at atmospheric pressure at elevated temperatures. In one instance, such a specimen was drawn to a reduction of area of 99.7 per cent, or an elongation at the neck of 300-fold, without fracture.

The picture of a material to which we could impart any specified strength by suitable cold working is a most intriguing one. There are, however, two hitches in carrying through such a rosy program. In the first place, the great strength is that under the pressure that imparts the ductility, whereas, of course, we are interested in the

(*Concluded on Page 220*)



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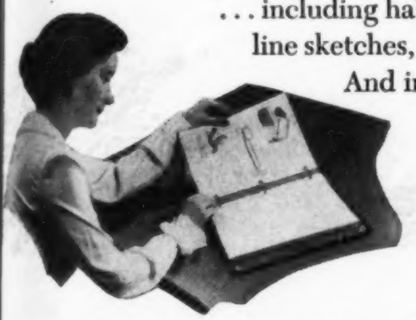

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(*Concluded from Page* 216)

strength under normal conditions of atmospheric pressure. This hitch, however, does not prove to be a serious one, because it is found experimentally that when the pressure is removed nearly all the strength under pressure is retained at atmospheric pressure. The second hitch is that the greatly enhanced strength is severely localized in the center of the neck where the maximum strain occurs, whereas we would like to have the strength distributed, as in a wire. If we could produce a wire of tensile strength 700,000 psi, which is a figure that has been realized at atmospheric pressure in necked specimens, we would have something that might be attractive industrially.

Draw Wire under Pressure

The suggestion naturally presents itself: why not conduct the entire wire drawing process under hydrostatic pressure, so that, by taking advantage of the great ductility imparted by pressure, the drawing process could be carried much further than normal before annealing, with greatly increased strain hardening and strength. The suggestion seems plausible, but the difficulty is in devising a practical method of doing it. Rather elaborate methods apparently would be necessary to conduct the drawing process under pressures as high as 400,000 psi, but the problem is not too formidable at lower pressures, and I have made experiments on wires drawn under a mean pressure of 180,000 psi. The results were in the expected direction, and wires were produced with strengths as high as 600,000 psi. However, these wires were so brittle that they could have little industrial application. It would appear that the reason that larger effects are not produced is that under normal drawing conditions the pressure in the throat of the die is already of the order of magnitude of 180,000 psi. In fact it is just this pressure in the throat of the die that makes possible the great elongations reached in ordinary wire drawing. If, by more elaborate means, wire drawing could be conducted at, say, twice the pressure, the brittleness would mostly disappear and we might anticipate alteration of properties of sufficient magnitude to be industrially significant.

From a paper entitled "Properties of Materials Under Superindustrial Stresses," the fifth Charles M. Schwab Memorial Lecture, delivered May 23, 1951 at the General Meeting of American Iron and Steel Institute in New York.

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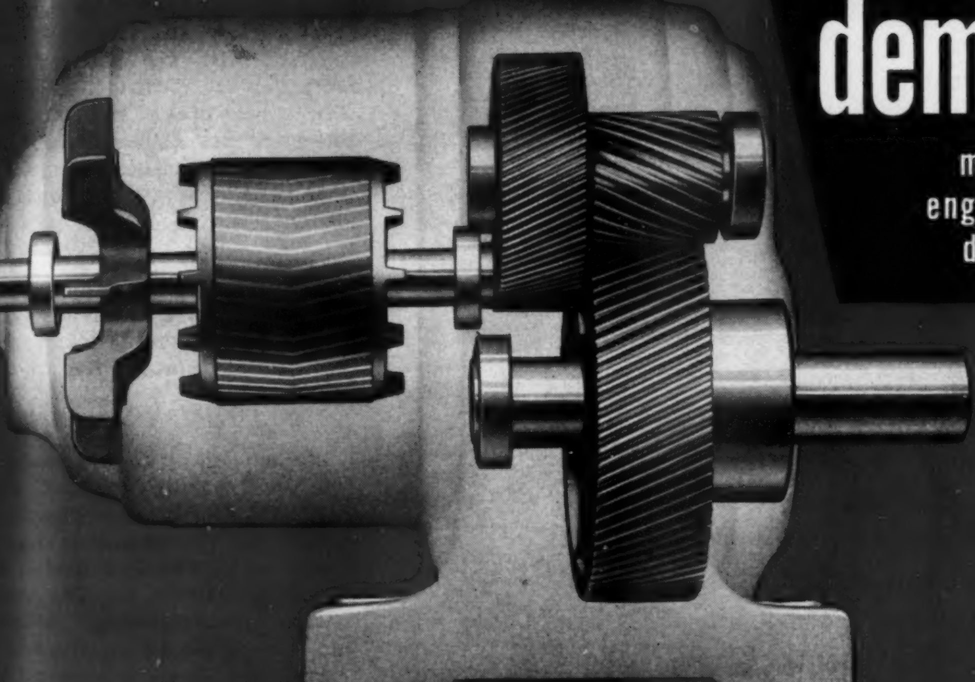

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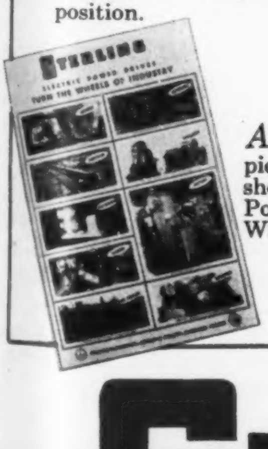

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News

OF MANUFACTURERS

Six manufacturing affiliates of the **General Electric Co.** became departments of the parent company on June 30. These affiliates are **Carboloy Co. Inc.**, with headquarters in Detroit; **General Electric X-Ray Corp.**, Milwaukee; **Locke Inc.**, Baltimore; **Telechron Inc.**, Ashland, Mass.; **Monowatt Inc.**, Providence, R. I.; and **The Trumbull Electric Manufacturing Co.**, Plainville, Conn.

♦

Barksdale Valves, manufacturer of extreme pressure Shear-Seal valves, has recently moved to its own plant at 1566 East Slauson Ave., Los Angeles, Calif.

♦

Necessitated by the increasing volume of both defense and civilian work, the second major plant expansion within the past year has been completed by **Precision Rubber Products Corp.**, Dayton, O. The additions include manufacturing, warehouse and office space which, with new production equipment, is expected to increase capacity by one-third.

♦

The sale of its industrial scale business to **Detecto Scales Inc.**, Brooklyn, N. Y., was announced recently by the Phliadelphia division of **Yale & Towne Manufacturing Co.**

♦

Allis-Chalmers Mfg. Co. has announced plans for the construction of a new plant for the manufacture of compressors for the Curtis-Wright Sapphire, an engine of 7200 lb thrust which powers the Republic F84F thunder jet fighter.

♦

The new $1,250,000 paint manufacturing plant of the **Pittsburgh Plate Glass Co.** was opened recently in Torrance, Calif. Located about 15 miles from downtown Los Angeles, the plant is equipped to produce a complete line of varnishes, resins, industrial and automotive finishes.

♦

Link-Belt Co. has started construction of a modern engineering and manufacturing plant for the production of elevating, conveying and processing machinery. The new plant is designed for efficient straight-line manufacture from the receiving department at one end of an 880-ft long

Avoid air power failure!

Choose Rivett Air Cylinders

The importance of smooth, steady cylinder performance cannot be over-emphasized in today's production. Only too often a whole line is slowed down, or stopped altogether because of a faulty cylinder.

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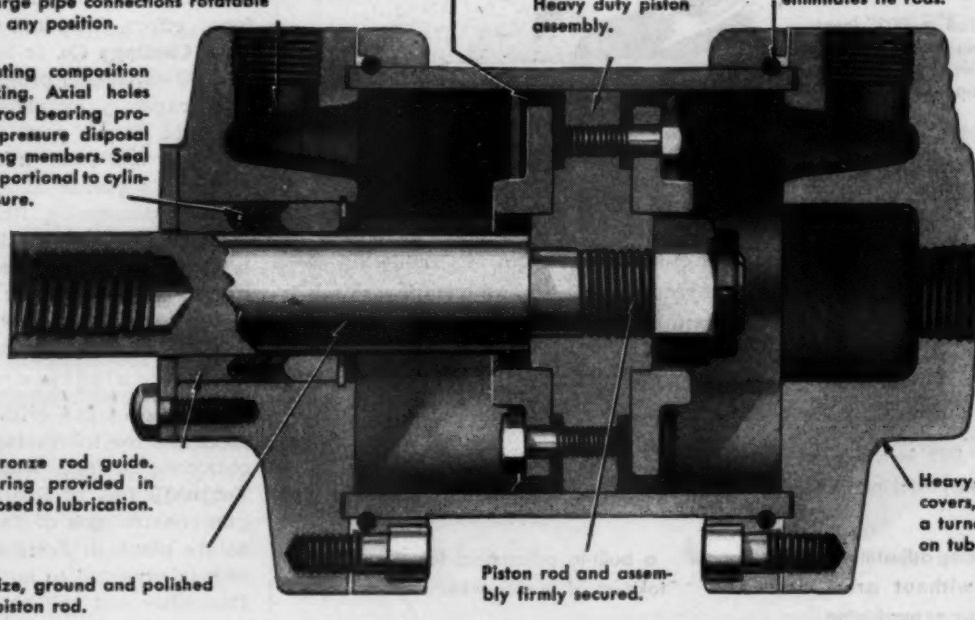
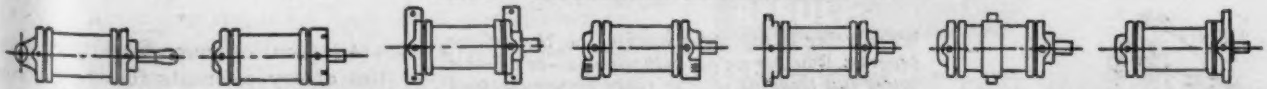
Recommended for pressures up to 150 P.S.I., Rivett air cylinders are available in 7 standard mountings; in 9 bore diameters, in any length stroke up to 96", single or double-end rods, with internal or external thread, and cushioned rod end, blind end, or both. Special covers supplied. Send today for Catalog Section 55.



SEVEN STANDARD MOUNTINGS



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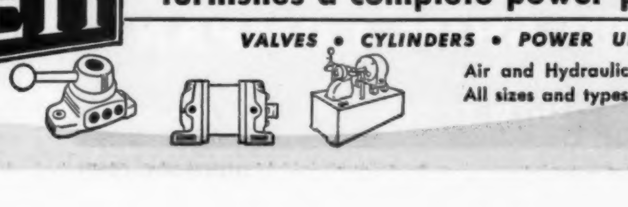


Air and Hydraulic
All sizes and types

building to the shipping department at the other end. Located on a 43-acre site at Colmar, Montgomery County, Pa., on the Doylestown branch of the Reading Railway, it will contain approximately 300,000 sq ft of floor space.

♦

Fabricast Division of **General Motors Corp.** will erect a plant for the manufacture of aluminum castings at Jones Mills, Ark. Construction will begin as soon as materials are available, and initial production will be predominantly on defense items. Permanent mold castings will be produced, many of them of the highly intricate type used in torque converter transmissions.

♦

A new **Westinghouse Electric Corp.** plant is now under construction in Union City, Ind. It will be geared to produce each month 75,000 electric motors ranging from 1/20 to ½ horsepower. Production is expected to begin before the end of the year.

♦

To increase production for the defense effort, **National Malleable and Steel Castings Co.** is beginning a $6,300,000 expansion program which will raise capacity by about 25 per cent with the larger part of the increase being in malleable iron. All of the company's plants at Cleveland, Chicago, Indianapolis, Sharon, Pa. and Melrose Park, Ill., will share in the improvements. The largest expenditures are planned for the Cleveland works, where an entire new malleable foundry unit will be added.

♦

Because of the critical shortage of steel and the increasing difficulties in obtaining it from Europe, **R. G. LeTourneau Inc.** of Peoria, Ill., has begun construction of its own steel mill at its plant in Longview, Tex. The mill is expected to be in operation by December and will be capable of producing 1000 tons per day of finished steel plate.

♦

Addition of manufacturing facilities of dry colorants for the coloring of crystal styrene and other thermoplastics has been announced by **Mid-America Plastics Inc.,** Cleveland. With years of coloring experience, the company has specialized in reprocessing plastic. Mid-America has purchased the rights and formulas of the dry color division of the **H. Jamison Co.,** New York City, and is producing dry colorants under the tradename of Colorblende, which is available in 17 colors matching the Bureau of Standard's colors.

Society

ACTIVITIES

At the annual meeting of the nominating committee, the **American Society for Metals** named the following officers and trustees to serve for the 1951-1952 term: President, Dr. John Chipman, Massachusetts Institute of Technology; vice president, Ralph L. Wilson, Timken Roller Bearing Co.; treasurer, Ralph L. Dowdell, University of Minnesota; and trustees, George A. Roberts, Vanadium-Alloys Steel Corp.; J. B. Johnson, Wright-Patterson Air Force Base; J. B. Austin, United States Steel Co.; Dr. James T. MacKenzie, American Cast Iron Pipe Co.; Walter E. Jominy, Chrysler Corp.; Dr. Chipman; Mr. Wilson; W. H. Eisenman; and Mr. Dowdell.

♦

Nelson E. Cook, general superintendent of galvanizing for Wheeling Steel Corp., Wheeling, W. Va., recently received the 1950 annual award of the Galvanizers' Committee, sponsored by the **American Zinc Institute**.

♦

B. N. Ashton, president of Electrol Inc., has been appointed chairman of the **Society of Automotive Engineers** A-12 committee, covering aircraft shock struts. This committee reviews and makes comments on military specifications related to shock struts and associated equipment and also works on industry problems arising from new developments.

♦

The Merit Award of the **American Society of Industrial Engineers** was presented recently to the United States Radiator Corp. for "leadership in research, engineering, design-styling and manufacture in the boiler and radiator field."

♦

At the annual meeting of the board of directors of the **American Iron and Steel Institute** the following officers were re-elected: President, Walter S. Tower; vice presidents, B. F. Fairless, president, United States Steel Corp., and Frank Purnell, chairman of The Youngstown Sheet and Tube Co.; and secretary, George S. Rose. Max D. Howell, vice president and treasurer of United States Steel Corp., was elected treasurer. At the 59th general meeting of the institute, Mr. Fairless received the Bessemer Medal

of the Iron and Steel Institute (British). **Wilbert G. Nichol** and **Walter N. Flanagan** were awarded the American Iron and Steel Institute Medal for 1950, and a newly established award, the Regional Meeting Technical Award, also for 1950, was presented to H. E. Warren Jr. These three men are also associated with United States Steel Co. At the same meeting Edward L. Ryerson, chairman, Inland Steel Co., Chicago, received the Gary Memorial Medal for outstanding achievement in the iron and steel industry.

♦

At the 35th annual meeting of the **American Gear Manufacturers Association** the Connell Award for 1951 was presented to Edward Whitney Miller, president of the Fellows Gear Shaper Co. This award was established by the Falk Corp. in commemoration of the late Edward P. Connell. Also at this meeting, held last month, the following officers were elected: President, George H. McBride, Westinghouse Electric Corp.; vice president, S. L. Crawshaw, Western Gear Works; treasurer, Louis B. Bond, Christiana Machine Co., who was re-elected to the position; and executive committee members, Marvin R. Anderson, Michigan Tool Co.; Gunnar E. Gunderson, Brad-Foote Gear Works Inc.; R. B. Holmes, Link-Belt Co.; and Charles R. Kessler, Beaver Gear Works.

♦

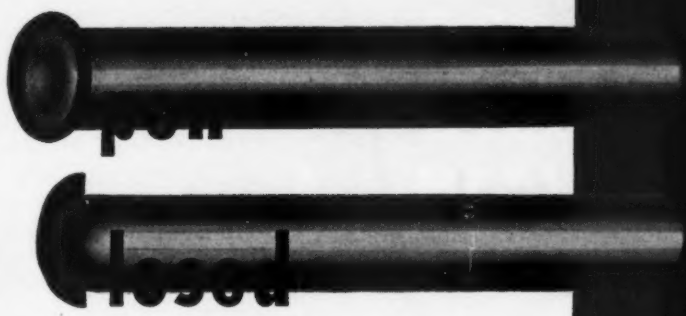
Fellow and past president of the **Industrial Designers Institute**, Ben Nash has been awarded the Medal for Achievement by the national board of the institute. The award was presented for his "courage and pioneering, for educational work and personal achievement, all in the field of Industrial Design."

♦

Following its annual practice, the **Society of Naval Architects and Marine Engineers** has awarded two graduate scholarships for the 1951-1952 academic year. Recipients of the new awards are Edward H. McCallig of Baltimore and Bjorn M. Olson of Cambridge, Mass.

♦

Gray Iron Founders' Society, Cleveland, has announced the opening of its 1951 redesign contest, which gives recognition to those who submit the best examples of the redesign of components for production in gray iron. The award program is open to all engineers, designers or other interested persons, whether or not they are employees of member companies. Prizes totaling $350 will be given to the top three winners.



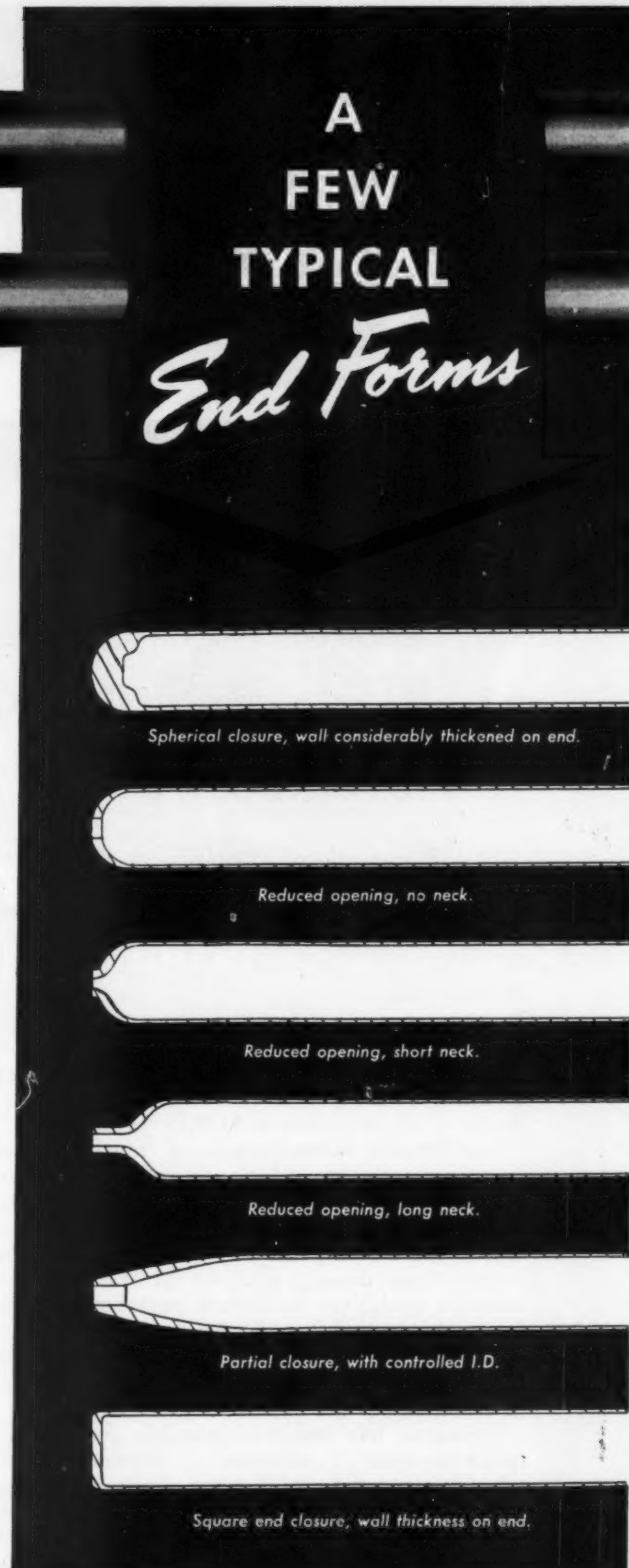
an Open and Closed case...

We employ this homely phrase because it suggests so well the idea of finality when applied to the Wolverine Spun End Process.* This process has solved so many problems, effecting economies and simplicity, that our reference as "an open and closed case" seems quite appropriate.

The process can be used most advantageously wherever the end of a tube is required to assume a rounded, tapered or related form—either completely or partially closed. Such end treatments often eliminate the use of additional parts and, of course, the attendant assemblies. They always tend to simplify the design and in most cases bring about worthwhile savings in material and labor.

Right now if you are contemplating production of tubular parts with special end treatments, consult our Customer Engineering Service. We are sure the help we can give you along this line will prove very valuable.

**a patented process RE: 22465*



WOLVERINE TUBE DIVISION
Calumet & Hecla Consolidated Copper Company
INCORPORATED
Manufacturers of seamless, non-ferrous tubing
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PLANTS IN DETROIT, MICHIGAN AND DECATUR, ALABAMA
Sales Offices in Principal Cities

Export Department, 13 E. 40th St., New York City 16, N. Y.

Personnel

ASSOCIATED with the Bristol Co., Waterbury, Conn., in sales work since 1943, **William Magenau** has been appointed field sales manager for the mill supply division of the company. Before coming to Bristol, Mr. Magenau was associated with the Page Belting Co. as a salesman and with the Dayton Rubber Co. as a sales engineer. He will make his headquarters at the company's main office in Waterbury.

•

Standard Pressed Steel Co., Jenkintown, Pa., recently announced two promotions in its sales force. **C. W. Hollingsworth** has been named divisional sales manager in charge of the Unbrako socket screw, Flexloc lock nut, and Hallowell steel collar divisions of the company. Succeeding Mr. Hollingsworth as manager of the Unbrako socket screw division is **Raymond N. Gruber.**

•

Appointment of **Willard deCamp Crater** as assistant sales manager of Marvinol vinyl resins for Naugatuck chemical division, United States Rubber Co., was announced recently. In his new capacity, Mr. Crater will develop new markets for vinyl resins and will direct sales activities. He will make his headquarters in the division's Naugatuck, Conn., plant.

•

Dudley B. Robinson has been appointed general sales manager of The Torrington Manufacturing Co., Torrington, Conn. Joining the company in May, 1949 as assistant general sales manager, Mr. Robinson brought to the company a background of sales experience gained in several industries.

•

Sterling Electric Motors Inc. has announced the following additions to its sales and engineering staff: **John F. Ingle** has been named district manager, with offices at 1066 Howard St., San Francisco, Calif., to serve central and northern California and western Nevada; **Robert P. Killion** is now a district manager serving southeastern Texas, with offices at 1213 Capitol Ave., Houston, Tex.; **H. L. Fritz,** 1667 Argonne Drive, Baltimore 18, Md., will serve the Maryland territory; and **Edmond W. Hodge Jr.,** 1727 Sixth Ave. North, Birming-

ham 3, Ala., will represent the company in the Alabama territory. **Joseph P. Foley, Charles E. O'Leary** and **Raymond S. Portner** have been added to the staff of the New York district office, 385 Gerard Ave., New York City; and **Melvin Maxham** and **John Malloy** are new members of the Los Angeles headquarters sales staff.

•

Anthony J. Zino Jr. has been appointed domestic general sales manager of Joseph Dixon Crucible Co., with headquarters in Jersey City, N. J.

•

Delta-Star Electric Co., division of H. K. Porter Co. Inc., has announced the appointment of **John Romano** as general sales manager. Mr. Romano has been with Delta-Star since 1927, starting in the engineering department and progressing through sales engineering to become assistant sales manager in 1945.

•

Robert L. Gibbs has been named manager of sales personnel of the Mueller Brass Co., Port Huron, Mich. Mr. Gibbs, who has been with the company for the past ten years, will be responsible for the direction of sales activties of its 26 district sales offices.

•

Sales manager for the American Phenolic Corp., Chicago, **William H. Rous** was elected vice president of the company at the annual meeting of the board of directors.

•

Russell J. Cameron, former vice president in charge of sales, has been elected president of Ross Operating Valve Co., Detroit. He succeeds **John Sainsbury,** who will remain as an active consultant and member of the board of directors.

•

George G. Raymond Jr. has been named executive vice president and general sales manager of the Lyon-Raymond Corp., Greene, N. Y.

•

Appointment of **Frank L. Brierly** as eastern division sales manager has been announced by the Townsend Co., New Brighton, Pa., manufacturers of rivets, bolts, nails and special fasteners. Having joined the company in 1931, Mr. Brierly's entire business experience has been in the fastener industry. His headquarters will now be in the company's Philadelphia office, and the territory which will be under his supervision includes New England, New York, New Jersey, Pennsylvania, Delaware, Maryland,

West Virginia and the District of Columbia. Industrial sales in Virginia and North Carolina also come under this division. Concurrently, announcement was made of the appointment of **Edward T. Brown** as special representative of the company, with headquarters in the general office at New Brighton.

♦

Robert A. Lees, manager of the Akron laboratories of American Anode Inc., has been loaned to the National Production Authority as an advisor on the allocation of liquid latex.

♦

The appointment of **Edward J. Brichta** as sales manager of the industrial air-cooled engine division of Continental Motors Corp. has been announced. Mr. Brichta has been with the company for 15 years, and in the sales department of the industrial air-cooled engine division since 1948.

♦

N. A. Guill has been named division manager of the Chiksan Co. Chicago branch office and sales supervisor of the Chicago district, with offices at 122 South Michigan Ave. He has been the company's sales representative in the Chicago territory for several years. **W. L. Clark** has been appointed sales representative in this territory and assistant to Mr. Guill. He joins the organization after many years with Crane Co. in the Chicago area.

♦

Formerly district manager for Machinery and Welder Corp., **John Fox** has been named representative of the welding products division of A. O. Smith Corp. in the greater St. Louis area.

♦

A number of new appointments have been announced recently by the General Electric Co. **S. Vernon Travis** has been made assistant general sales manager of the company's large apparatus division at Schenectady, N. Y. He is succeeded in his former position as manager of sales for the large motor and generator divisions by **Louis H. Matthes. B. R. McClure** was recently appointed assistant to the manager of sales of the fractional horsepower motor divisions at Fort Wayne, Ind., and, at the same time, **D. C. Hanson** was made manager of the refrigeration equipment sales division and **K. R. Whearley** was placed in charge of the distribution, parts and service sales section. **Richard S. Walsh** has been named manager of the induction motor sales di-

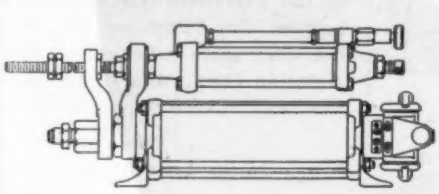
Built-in Controls Make This AIR CYLINDER POWER UNIT

Fast, Smooth, Compact

Model HCBEM5B-60

TWO MOUNTING STYLES

Model HCBEM Air Motor and Hydro-Check is available arranged in tandem, as shown above, or in parallel, as sketched below:



Five bore sizes: 1¼", 1¾", 2½", 3⅜" and 4½". Stroke lengths up to 18" or more. Combination units of Air Motor and Hydro-Check are also made equipped with the standard Bellows manually operated valve; and of course, Air Motors and Hydro-Checks are available as separate units.

You get the smooth piston travel characteristic of hydraulic operation plus the fast flexibility of air power from the Bellows Air Motor and Hydro-Check, combined in a single unit.

This compact power unit has the built-in ELECTRO-AIRE* directional valve; electrically controlled, but *air-powered.* The valve operates on 8 volts, so you can use simple low voltage wiring. No electrical hazard to operator or machine. Place your controls where you wish. The piston rod responds instantly, without lag or delay.

This power unit is fast. The valve can make as many as 2200 movements an hour, all day long, without overheating. Built-in speed controls (one for the power stroke, one for the retract) let you adjust piston rod speed to your own needs.

But the Hydro-Check is the control element that makes this power unit so radically different. The Hydro-Check flattens out the bounce or chatter that occurs as a result of the natural compressibility of air. The air cylinder piston movement becomes miraculously smooth. You have the smoothness of hydraulic operation plus the advantage of air's speed and flexibility. You can set the Hydro-Check to take over control at any point; so you can have rapid advance and rapid return, yet have full control over the feeding stroke.

All in all you just can't beat the combination of Bellows Air Motor and Hydro-Check for fast, smooth, accurately controlled air cylinder power.

* TM Reg. Patent Pending

Write for this FREE Bulletin

Photographs, wiring diagrams, technical data, case histories, etc., on Bellows Air Motors, Hydro-Checks and other Bellows "Controlled-Air-Power" Devices. Shows how easy it is to use flexible, economical air power for "Faster, Safer, Better, Production." Address Dept. MD751, The Bellows Co., Akron, 9, Ohio.

1255

The Bellows Co.
AKRON 9, OHIO

vision of the small and medium motor divisions at Schenectady; **C. B. Seelig** was appointed assistant manager of the Fitchburg, Mass., turbine sales division of the turbine divisions; and **E. E. Hinson** of the chemical department has been assigned as sales representative in Washington, D. C. He will be located at 806 Fifteenth St., N. W., and will have responsibility for the sale of industrial laminated plastics and electric insulating materials.

•

Joseph L. O'Brien, sales manager of the Doehler-Jarvis Corp., Chicago division, has been appointed consultant to the Office of Price Stabilization, division of the Economic Stabilization Agency. **William L. Huber,** formerly a member of the sales staff of the company's Chicago division, succeeds Mr. O'Brien as sales manager.

•

The Osborn Manufacturing Co., Cleveland, has appointed **A. J. Steffens Jr.** as sales engineer, covering the territory of North Carolina, South Carolina, eastern Georgia and Florida.

•

Alan Cameron, 25-year veteran in aircraft supply and technical sales, has been named manager of the newly created foundry division of Rosan Inc., South Gate, Calif. Mr. Cameron has served as special representative of the Firestone Aircraft Co., subsidiary of the Firestone Tire and Rubber Co.; sales manager of the Scott Aviation Corp.; and sales manager of the Trader Aero Supply of Pittsburgh.

•

Associated with Graver Tank and Mfg. Co. Inc. for more than two years as manager of sales, first for the weldment division and later for the alloy division as well, **Harry A. Dennis** has been appointed assistant sales manager for the company.

•

Ralph W. Burk, vice president of sales for Kearney & Trecker Corp. since 1943, was recently elected vice president for manufacturing. He will also continue as head of the sales division.

•

Changes involving personnel of the factory service organization of Cummins Engine Co. Inc., Columbus, Ind., were announced recently. **Charles C. Sons** has been appointed acting eastern service manager, with headquarters in Columbus, and **Dillard B. Davis,** formerly eastern service manager, is now regional service representative in the central region. He

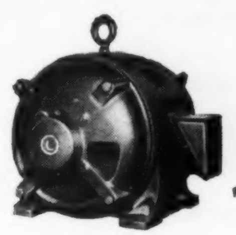
Guard Against Atmospheric Hazards...

WITH

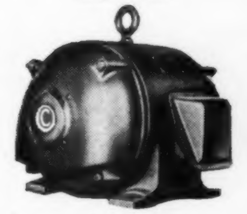
Century

PROTECTED MOTORS

Century Electric Company is celebrating its 50th year in the electrical industry.

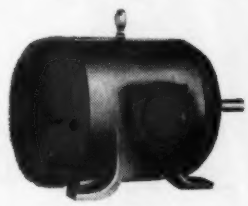


Drip Proof



Splash Proof

Totally Enclosed Fan Cooled



Explosion Proof

Century C MOTORS

ALTERNATING CURRENT MOTORS

POLYPHASE

- Squirrel Cage Induction — 1/6 to 400 H.P.
- Wound Rotor Motors—1 to 400 H.P.
- Synchronous Motors—20 to 150 H.P.

SINGLE PHASE

- Split Phase Induction—1/6, 1/4, 1/3 H.P.
- Capacitor—1/6 to 20 H.P.
- Repulsion Start, Brush Lifting, Induction—1/2 to 20 H.P.

DIRECT CURRENT MOTORS

1/6 to 300 H.P.

GENERATORS

AC, .63 to 250 KVA
DC, .75 to 200 KW

GEAR MOTORS

1/8 to 1-1/2 H.P.

MOTOR GENERATOR SETS

AC to DC, AC to AC
DC to DC, DC to AC

Open Protected, Splash Proof, Totally Enclosed Fan Cooled, Explosion Proof.

Ball Bearing motors are factory lubricated for several years' normal service. Bearing housing construction permits easy re-lubrication when unusual service demands it.

CE-673

To guard your production against the destructive effects of atmospheric hazards, Century offers four types of protective motor frames.

DRIP PROOF—meets the requirements of most installations. Use it where operating conditions are relatively clean and dry. Top half of the frame is enclosed to keep out falling solids and dripping liquids.

SPLASH PROOF—keeps splashing liquids out of the motor even when the frame is washed with the full force of a hose. Use Century Splash Proof motors indoors or outdoors.

TOTALLY ENCLOSED FAN COOLED—resists the hazards of abnormal concentrations of dusts, powders, grit, oil mists, acid and alkali fumes.

EXPLOSION PROOF—protects life and property in atmospheres charged with explosive dusts or vapors.

The properly selected protection with the wide variation of starting torque characteristics to choose from provides long operating life and improves the production of the driven equipment.

Century motors are available in a wide range of kinds and types—in sizes from 1/6 to 400 horsepower—for single phase, polyphase and direct current applications.

Specify Century motors for all your electric power requirements.

CENTURY ELECTRIC CO. 1806 Pine St. • St. Louis 3, Mo.

Offices and Stock Points in Principal Cities

makes his headquarters in Chicago and replaces **Lloyd Kerber,** who resigned to accept the position of general service manager for Cummins Diesel Sales Corp. of Missouri, at St. Louis.

♦

R. A. Gloss has been appointed sales representative for the Wisconsin and upper peninsula Michigan territory of Federated Metals division, American Smelting and Refining Co. He will work out of the company's district sales office at 756 North Milwaukee St., Milwaukee, Wis.

♦

The Plaskon division of Libbey-Owens-Ford Glass Co. has announced that **Donald O. Wynocker** has been named to assist **Charles Walker** in a newly expanded sales area for the company's glues and industrial resins. Mr. Wynocker will assist in covering Mr. Walker's present territory of Ohio, Michigan and Indiana, which will be extended to include Pennsylvania, New York and the New England states.

♦

Joseph Markowski, formerly industrial engineer for Consolidated Vultee Aircraft Corp., has been appointed assistant managing field engineer for The Work-Factor Co., New York.

♦

Rockwell Manufacturing Co., Pittsburgh, has announced the appointment of **Norman W. Rowand** as general manager of the Pittsburgh plant of the company's meter and valve division. He was formerly chief industrial engineer for the Pittsburgh-DuBois division, DuBois, Pa., and plant manager of the regulator division at Norwalk, O.

♦

Edward N. Case has been appointed product supervisor for metal trades product sales in the synthetic organic chemicals department, industrial chemicals division of American Cyanamid Co. He will assume responsibility for the marketing of all of the department's metal trades products and will make his headquarters at the company's New York office.

♦

Appointment of **C. W. Powell** as Pittsburgh branch manager has been announced by Carboloy Co. Inc., Detroit. Mr. Powell, who will be located at 704 Second Ave., Pittsburgh 19, Pa., joined the company in 1943 as sales and service manager in both the midwestern and east central districts.

SALES Notes

RECENTLY announced was the appointment of **Van Dusen Aircraft Supplies Inc.,** 2004 Lyndale Ave., Minneapolis, Minn., as distributor for precision O-rings manufactured by **The Parker Appliance Co.,** Cleveland. Presently a distributor of Parker aircraft valves, fittings, and other parts, the company will now maintain complete service stocks of O-rings.

♦

Facilities of the **Industrial Control Co.** have been moved from 26-02 Fourth St., Long Island City 2, N. Y., to a new location on Straight Path and Arlington Ave., Wyandanch, L. I., N. Y.

♦

Mechanical Air Controls Inc. has moved into a new building at 15311 West Eleven Mile Rd., Royal Oak, Mich. New models of MAC air valves are in production to complete a full line of all types in the ¼ to ¾-in. sizes, and engineering and tooling is under way to add pilot-operated models in larger sizes, as well as valves of special design. The new plant facilities will assure prompt deliveries.

♦

Two **Waldes Kohinoor Inc.** Truarc retaining ring distributors and one representative have announced removal to larger quarters. **Ehret & Kinsey,** Chicago representative, has moved to 141 West Jackson Blvd.; **Bobker Bearings,** New York and New Jersey distributor, has moved to 282 Seventh Ave., New York City; and **Bearings Service and Supply,** Denver distributor, is now located at 1850 Market St.

♦

Organized for the manufacture of high-strength glass-reinforced plastics parts, **The Dynakon Corp.,** 5509 Hough Ave., Cleveland 3, O., has taken over the operations of the plastics products division of **The Central Electrotype Co.** of Cleveland. The new organization will specialize in the engineering and manufacture of glass-fiber reinforced plastics parts whose strength, electrical and chemical properties have been of value to the mechanical, chemical, electrical, and plating industries. Patented processes of the corporation are being made available under license to qualified manufacturers.

Why monkey around with inferior tubing?

Bundyweld Tubing, double-walled from a single strip. Exclusive, patented beveled edge affords smoother joint, absence of bead, less chance for any leakage.

Why fuss around with the costly production delays, poor performance and other monkey business an inadequate tubing can get you into?

Let Bundyweld show you what tubing features really are!

This multiple-wall type of Bundy® tubing is double-rolled from a single strip. No other like it. It's extra-rugged, easy to form and highly resistant to vibration fatigue. It's thinner walled, yet stronger walled, won't leak under pressure or burst under normal strain.

For help on any application of small-diameter tubing, why not check Bundy Tubing Company today?

Bundy Tubing Company

DETROIT 14, MICHIGAN

World's largest producer of small-diameter tubing

AFFILIATED PLANTS IN ENGLAND, FRANCE AND GERMANY

Meetings

AND EXPOSITIONS

July 30-Aug. 2—
American Electroplaters' Society. 38th annual meeting to be held at the Statler Hotel, Buffalo, N. Y. Additional information is obtainable from American Electroplaters' Society, 473 York Rd., Jenkintown, Pa.

Aug. 13-15—
Society of Automotive Engineers. West Coast meeting to be held at the Olympic Hotel, Seattle, Wash. John A. C. Warner, 29 West 39th St., New York 18, N. Y., is secretary and general manager.

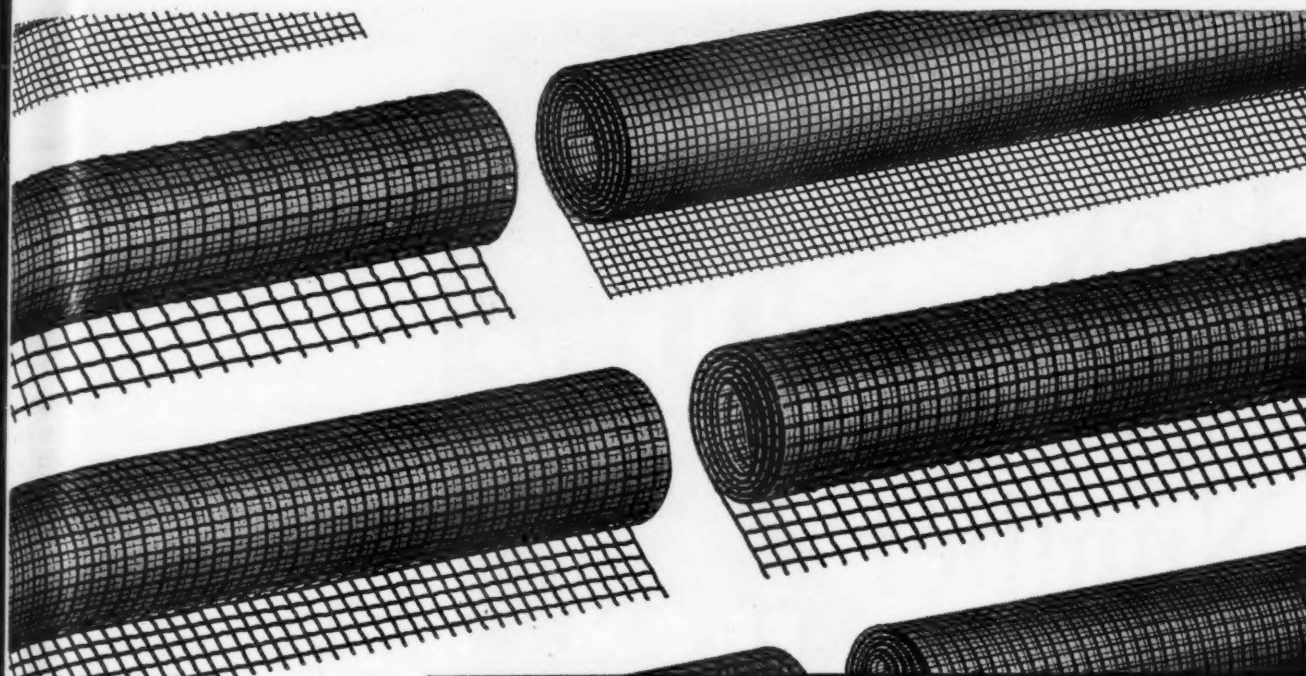
Aug. 30-Sept. 13—
The British Engineers Association. Engineering, Marine and Welding Exhibition to be held in Olympia, London, England. Additional information may be obtained from J. V. Topham, secretary, 32 Victoria St., London, S. W. 1., England.

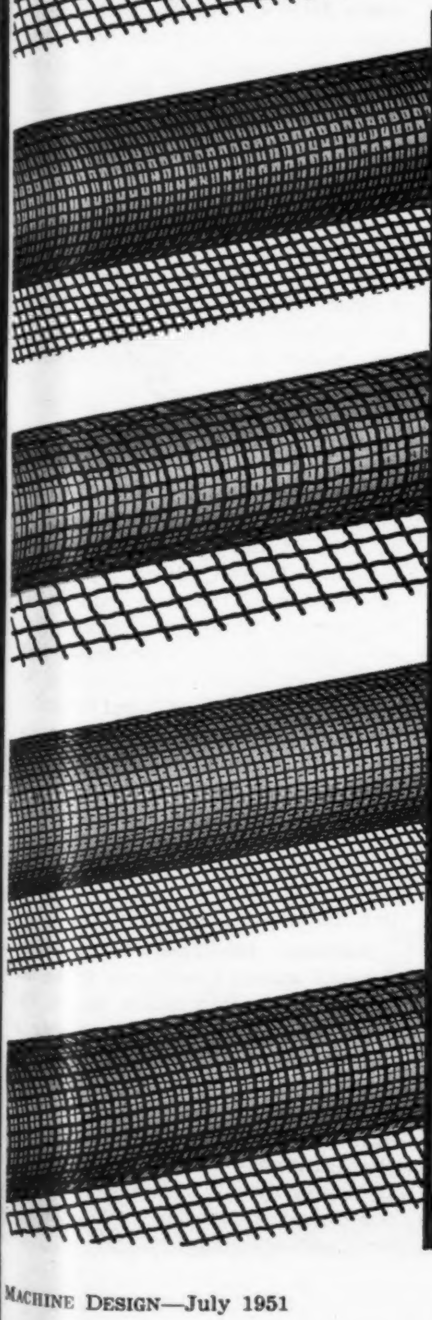
Sept. 1-10—
European Machine Tool Exhibitions. First exhibition to be held at the Porte de Versailles, Paris, France. Additional information may be obtained from A. J. Gibbs Smith, 477 Streetsbrook Rd., Solihull (Warwickshire) England.

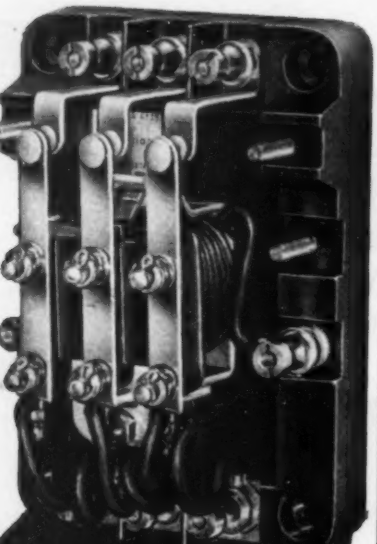
Sept. 3-7—
Institute of the Aeronautical Sciences — Royal Aeronautical Society. Third International Aeronautical Conference to be held at Brighton, Sussex, England. Additional information may be obtained from Robert R. Dexter, secretary, 2 East 64th St., New York 21, N. Y.

Sept. 10-13—
Society of Automotive Engineers. Tractor and production forum to be held at the Schroeder Hotel, Milwaukee, Wis. John A. C. Warner, 29 West 39th St., New York 18, N. Y., is secretary and general manager. (Headquarters for this meeting were incorrectly listed in our June issue as being at the Biltmore Hotel, Los Angeles, Calif.)

Sept. 10-14—
Instrument Society of America. Sixth National Instrument Conference and Exhibit to be held in the Sam Houston Coliseum, Houston, Texas. Additional information may be ob-







tained from society headquarters, 921 Ridge Ave., Pittsburgh 12, Pa.

Sept. 10-14—

American Society of Mechanical Engineers. Industrial Instruments and Regulators Division, joint conference with Instrument Society of America to be held at Houston, Texas. C. E. Davies, 29 West 39th St., New York 18, N. Y. is secretary.

Sept. 24-26—

American Society of Mechanical Engineers. Petroleum mechanical engineering conference to be held at Hotel Mayo, Tulsa, Okla. C. E. Davies, 29 West 39th St., New York, N. Y., is secretary.

Oct. 11-12—

ASME Fuels and AIME Coal Division Joint Conference to be held at Hotel Roanoke, Roanoke, Va. Additional information may be obtained from C. E. Davies, 29 West 39th St., New York, N. Y.

Oct. 14-19—

American Welding Society. Thirty second annual meeting to be held at Hotel Book Cadillac, Detroit, Mich. Additional information may be obtained from society headquarters at the Hotel Book Cadillac, Detroit, Mich.

Oct. 15-19—

American Society for Metals. Thirty third annual metal show to be held at the Michigan State Fair Grounds. Detroit, Mich., under the sponsorship of the American Society for Metals; American Welding Society; Metals Branch, American Institute of Mining and Metallurgical Engineers; Society for Non-Destructive Testing. Additional information may be obtained from society headquarters, National Metal Congress & Exposition, 7301 Euclid Ave., Cleveland 3, O.

Oct. 22-24—

National Electronics Conference. Seventh annual conference to be held at the Edgewater Beach Hotel, Chicago, Ill., under the sponsorship of the American Institute of Electrical Engineers, the Institute of Radio Engineers, Illinois Institute of Technology, Northwestern University, and the University of Illinois, with participation by the University of Wisconsin and the Society of Motion Picture and Television Engineers.

Stop High-Pressure Leaks and Blow-offs

—with ANCHOR Ductile Sleeve Hose Couplings

Factory application of Anchor's exclusive Ductile Sleeve Coupling is your best insurance against leaks or blow-offs. Our superior grip gives you a plus margin of safety and a minimum time loss due to lay-up of equipment.

Here's how it works (referring to the illustration): Cover stock (G) is removed from the hose by grinding and buffing. Ductile Sleeve (C) is slipped on the end of the hose over the bare wire. Coupling body (A) with steel insert (B) attached is placed in position on the hose. The coupling is then swaged radially on the hose, imbedding the ductile sleeve into the internal grooves of the coupling shell (E) and into the mesh of the wire braid reinforcement (F). This is the exclusive Anchor patented grip.

Some of the applications of Anchor high-pressure flexible oil lines include high-pressure riveting, road machinery, snow plows, coal mining machinery, material handling machinery, machine tool applications, agriculture, railroads, oil field machinery, liquefied petroleum gas, lubrication and various other installations.

Clamp-type Couplings

2-Piece Re-usable Couplings

NPTF Straight Adapter Unions

45°-90° Adapter Unions

Other styles of related fittings available for piping installations

ANCHOR COUPLING CO. INC.

Factory: Libertyville, Illinois • Branch: Detroit, Michigan

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Dept. MD71, Libertyville, Illinois

I want engineered-information on cost-cutting Anchor Ductile Sleeve Hose Couplings. Please send me Bulletin No. 48.

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D-1

New Machines

Business Equipment

ACCOUNTING MACHINE: Model E machine provides automatic computation and printing of debit or credit balances, automatic proof total and, when necessary, automatic adjustment of proof total. Has two crossfooters for accounting application flexibility. Ten key machine features automatic column selection, front-feed carriage with writing line and all other entries visible, automatic carriage open and close, and printing of month, day, year and descriptive characters. In 18 and 24-in. carriage widths. *Underwood Corp., New York, N. Y.*

Heating and Ventilating

DEHUMIDIFIER: Model DMS 4 will dehumidify any tight closed area up to 8000 cu ft. Air drawn into unit by fan, passed over cold coils to release water which drops into pan for removal. Specifications: hermetically-sealed compressor; air flow, 110-cfm; dimensions, 17 by 13 by 15½ in.; weight, 55 lb. *Abbeon Supply Co., Woodside, N. Y.*

AIR CONDITIONER: Window type unit for room cooling. Front consists of five plastic parts across 12-in. high and 26-in. long front of unit. Three louver disks in front can be rotated to give desired air flow. *General Electric Co., Pittsfield, Mass.*

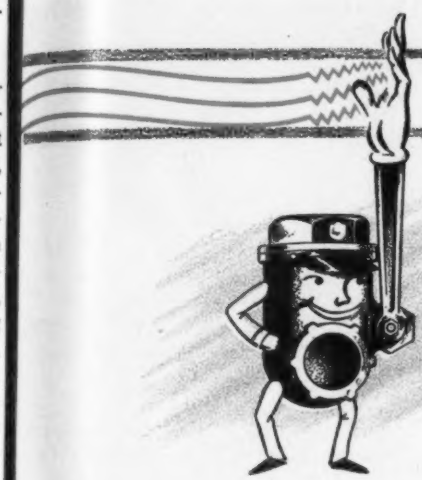
Manufacturing

WELDING MACHINE: New UE-48 machine for uniform welding of straight-line seams in light to heavy plate. Consists of side beam carriage, welding head, rod reel, feed hopper and controls. Welding current, carriage speed, welding voltage and welding material flow are adjusted before welding is begun. Electronic governor controls speed of carriage from 3 to 200 in. per minute. Welds metal from 16-gage to 1½-in. thick in one pass. *Linde Air Products Co., New York, N.Y.*

METAL DISINTEGRATORS: Line of disintegrators featuring automatic control without tubes. Current to operating head rectified to d-c. Head constructed so contact is made 60 times per second at a-c peak of half sine wave. Solenoid actuates spindle downward to assure contact with metal at all times. Typical average cutting speeds—disintegrates ⅛-in. drill, 2 in. deep, in

Specify this Fast-action Gate that seats tight and stays tight

JENKINS



BRONZE GATE VALVE

125 lbs. Steam
200 lbs. O.W.G.
½" to 2"

Wherever full, free flow is essential . . . where valve opening and closing must be instantaneous . . . Jenkins SWINGTITE Fast-Action Bronze Gate Valves are setting new standards of performance and endurance.

The exclusive rolling disc and guide track design in the new Jenkins SWINGTITE distributes the wear, assuring maximum tightness (since it prevents uneven wear of seating surfaces), and lengthens valve life.

If you design any equipment requiring Fast-Action Gates, find out how Jenkins can help you make your product more efficient and dependable. Send for folder describing the new SWINGTITE Bronze Lever Gate Valve. You add sales-appeal to your product, too, when you specify the SWINGTITE. It's made by Jenkins Bros., a name your customers know and trust. Write today for Form No. 196. Jenkins Bros., 100 Park Ave., New York 17; Jenkins Bros., Ltd., Montreal.

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- Dish-washing equipment
- Gasoline and fuel oil lines or motors, burners, etc.
- Fire-extinguishing steam lines in kitchens

AND FOR APPARATUS DESIGNED FOR USE IN...

- Oil refineries
- Textile finishing plants
- Chemical and food plants
- Pulp and paper mills

ONLY THE SWINGTITE HAS IT—

ROLLER ACTION

As the valve is opened or closed, guide rims (A) around the seating surfaces of discs roll freely over guide tracks (B) cast in the body, distributing wear evenly, dislodging foreign matter, and providing a polishing action for seating surfaces. This rolling disc and track construction lengthens valve life and assures maximum tightness.

JENKINS VALVES

LOOK FOR THE DIAMOND MARK

SINCE 1864 — TRADE JENKINS MARK

Jenkins Bros.

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1½ minutes. *Electro Arc Manufacturing Co., Detroit, Mich.*

WELDING HEAD: For welding dissimilar metals in assembly of radio and other electronic parts and in assembly of other small parts. Bench-mounted, press type unit uses single post mounting. Electrode pressure maintained through closed air system using metal bellows. Quick electrode follow-up achieved by keeping mass of upper electrode small. Model I-S handles ⅛ or ¼-in. electrodes. Opening between faces of upper and lower electrodes adjustable from 1 to 4 in. Dimensions; 3½ in. wide, 9½ in. high, 8½ in. deep. *Raytheon Manufacturing Co., Waltham, Mass.*

PORTABLE METAL SLITTER: Models 2024 and 1624 metal slitting machines handle up to 16 or 20 gage mild steel, respectively. Applications include sheets or coils for gutters, flashings, downspouts, flue pipe, etc. When ripping lighter gages, two or more sheets can be cut at same time. Throat depth of 27 in. permits cutting to center of 4-ft sheets of any length. *Wilder Manufacturing Co., Monterey, Calif.*

WELDING HEAD: Double-head welding head for small parts. Pressure is applied through cam acting against variable deflection of cantilever spring. Combination of cam and spring arranged to provide automatic follow-through pressure instantaneously. Independent switches and treadles for each pair of electrodes. Electrode pressure variable from 6 oz to 15 lb by micrometer type adjustment. *Federal Tool Engineering Co., Newark, N. J.*

DRILLING MACHINES: Vertical hydraulic drilling machines in 3, 5, 10 and 15-hp models. Geared heads mounted on top of columns, providing range of speeds through pick-off gears. Barnes hydraulic type 34 feed and traverse circuit used to actuate sliding head, with feeds from ½ to 12 in. per minute available. Dwell obtained by setting valve; skip feed and step drilling can be included. Three-hp machine has 12-in. stroke, drill capacity of 1½ in. in steel at 0.010-in. feed per revolution; 5-hp machine has 15-in. stroke, capacity of 2½ in. in steel at same feed per revolution. *Standard Machine & Tool Co. Ltd., Windsor, Ont.*

SPRING COILER: Hand-operated spring coiling machine for making compression, extension or torsion springs. Especially designed for experimental or sample springs or for production runs up to 500 springs. Setup time, usually less than 5 minutes; capacity, 300

springs per hour. Compression springs wound to 8 in. long with any number of coils to 30, any pitch to 1 in., with end coils closed or squared. Extension or coil springs wound tight or loose, left or right hand, up to 150 coils with extended arms on both ends to 5 in. long. Max outside diam, 2 in.; wire diam range, 0.041 to 0.156-in. *The Carlson Co., New York, N. Y.*

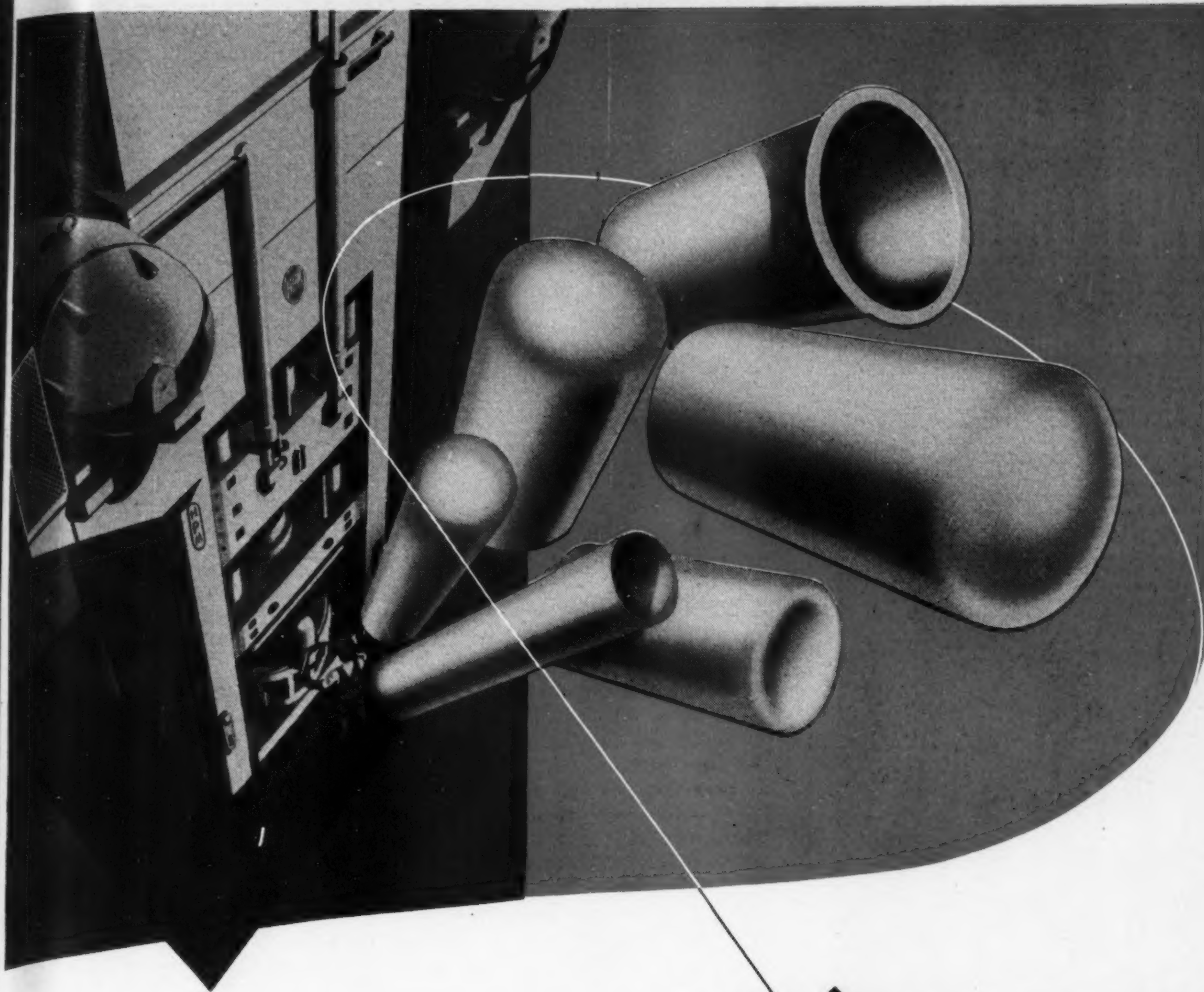
GEAR-HOBBING MACHINE: Semiautomatic machine for cutting gears with straight teeth, with teeth inclined up to 18 degrees, worm wheels, and bevel gears (with attachment). Lambert Type 75 machine has automatic radial and longitudinal feed which combine to shorten overall stroke. Minimum diam of gears which can be cut, $\frac{3}{64}$-in.; max diam, 3.47 in.; max length hobbed for straight gears, 1.969 in., for gears with inclined teeth, 1.181 in. *Carl Hirschmann Co., Manhasset, N. Y.*

WET-BLASTER: Low-cost bench model Cro-Hone Jr. wet blasting machine for cleaning metal parts. Uses fine abrasive suspended in water carrier and forced against parts by air pressure through siphon jet gun. Unit has no moving parts or circulating pumps. Controls located within reach of operator; large window permits watching work. Requires only air and water connections. *The Cro-Plate Co. Inc., Hartford, Conn.*

INJECTION MOLDER: New 8-oz L-2-8 injection molding machine combines high speed with rigid locking. Uses double-toggle linkage, more efficient injection cylinder design. Molds 8 oz of polystyrene material or 10 oz of acetate with effective plunger displacement of 32 cu in. Locking pressure, 350 tons; die plates, 22⅛ by 28 in.; injection pressure, 20,000 psi. *Lester-Phoenix Inc., Cleveland, O.*

ELECTRIC HAND DRILL: Portable drill geared down to 450 rpm for drilling high tensile steel, brick, glass, etc. Has capacity of $\frac{5}{16}$-in. in steel, ⅝-in. in wood, weighs 3 lb. Silicon alloy pressure castings used throughout, with nickel-chromium molybdenum steel helical gears. Desoutter drill is designed for one-hand operation. *Newage International Inc., New York, N. Y.*

BAND SAW TABLE: Capacity, 3000 lb. Measures 40 by 48 in. with 36 by 87-in. carriage bed. Bed connects to machine frame, is supported on steel legs. Table surface is 45⅞ in. from floor, has 18-fpm hydraulic feed stroke, 36-fpm reverse stroke. Includes 5-gal reservoir and



Are you interested in Deep-Drawn Shapes that are **strong** and light?

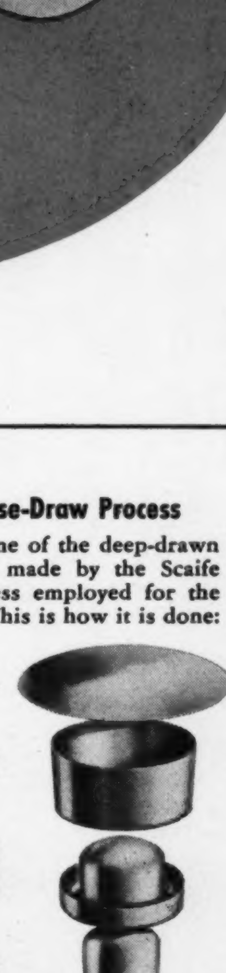
Here is a modern, proven method of making deep-drawn shapes that avoids conditions that produce thin, weak wall sections, or thick, heavy sections. Wall thicknesses produced by the Scaife method are uniform throughout the entire length of the draw.

Various symmetrical shapes can be produced from both ferrous and non-ferrous metals, in one continuous stroke of the press by this method. Representative applications include pressure vessels, containers, protective enclosures and many other parts.

We will be glad to have your inquiries regarding deep-drawn shapes up to 36 inches in diameter and ¼-inch wall thickness.

The Scaife Reverse-Draw Process

Shown above are some of the deep-drawn shapes that may be made by the Scaife process, and the press employed for the pressing operation. This is how it is done:



Starting with a circular sheet of steel—

a cup is formed by a conventional drawing operation.

A continuation of this pressing operation turns the cup "inside out" without removing it from the dies—

completing—in a single stroke—the deep-drawn shape.

FOUNDED 1802

Scaife Company

Founded 1802

Oakmont (Pittsburgh District) Penna.

2-gpm hydraulic pump driven by ¾-hp motor. *The DoAll Co., Des Plaines, Ill.*

TUMBLING MACHINE: For deburring and smoothing rough edges and sharp corners of steel, diecast or other metal parts. Easily replaceable linings reduce down-time required for relining. Semisoft rubber linings available. Variable-speed drive gives choice of tumbling speeds. *Crown Rheostat & Supply Co., Chicago, Ill.*

CUT-OFF MACHINE: Model M-14 metal cut-off machine for low-cost cutting of ferrous and nonferrous metals at rate of 4 seconds per sq in. Capacity, 1½-in. solids, 2½-in. pipe and structurals, and miscellaneous cutting of 2-in. solids. Uses 3½-hp geared-in-head motor. Swivels up to 45 degrees in either direction, requires bench space of 32 by 34 in.; weighs 300 lb. *Stone Machinery Co. Inc., Manlius, N. Y.*

BEAM PUNCH: Guillotine beam punch for punching flanges and webs of beams from 6 to 30 in. Deflection is straight up, does not cause buckling effect on punch and die units. No. 9 punch has 200-ton capacity, will punch four $\frac{13}{16}$-in. diameter holes through $\frac{11}{16}$-in. mild steel plate. Additional holes punched by staggering lengths of punch stems. *Beatty Machine & Manufacturing Co., Hammond, Ind.*

DRILL UNIT: Air-powered model 19-400 hydraulic drill unit for drilling, reaming, tapping, etc. Thrust obtained from plant air supply; controlled by sealed hydraulic system. Unit driven by electric motor or by belt. Length of approach and total stroke controlled in 0.005-in. steps at 5 in. per second rapid traverse, final depth controlled to 0.0005-in. Feed infinitely variable between zero and 50 in. per minute. Full stroke is 4 in. Requires ¾ to 1½-hp motor, handles up to ⅝-in. drill in steel. *Delta Power Tool Div., Rockwell Mfg. Co., Milwaukee, Wis.*

NIBBLING MACHINE: Newly designed Worcester nibbler features 2-speed drive and variable stroke adjustment to accommodate different thicknesses of metal. Permits straight or contour cutting at 1200 punch strokes per minute. One model handles up to $\frac{3}{16}$-in. low carbon steel; other model designed for up to $\frac{1}{16}$-in. low carbon steel. *Arduini Manufacturing Corp., Worcester, Mass.*

ELECTRIC HAND DRILL: Heavy-duty ½-in. drill for severe service in plants, shops, garages, etc. Capac-

ity in metal, ½-in.; capacity in hard wood, 1½ in.; no-load speed, 550 rpm; universal motor, a-c/d-c up to 60 cycles; length overall, 13⅝-in.; weight, 8½ lb. *Cummins Portable Tools, Division of Cummins-Chicago Corp., Chicago, Ill.*

Materials Handling

TROLLEY HOIST: In 5 and 7½-ton capacities. Low headroom design features welded steel trolley frame and hoist gear housing. Available with mechanical load brake of Weston screw type or with dynamic lowering using a-c or d-c controls. *Conco Engineering Works, Mendota, Ill.*

FORK TRUCK: Capacity, 2000 lb load, 48-in. long. Dimensions: width, 32½ in.; height, 83 in.; telescoping lift, 130½ in. loaded; initial lift, 65 in. loaded. Characteristics: outside turning radius, 65 in.; minimum intersecting aisles, 59 in.; right-angle turn, 79½ in. plus length of load; weight with battery, 4950 lb. *Baker Industrial Truck Div., The Baker-Raulang Co., Cleveland, O.*

ELECTRIC FORK TRUCK: Capacity, 6000 lb. Features single-cylinder, low-pressure, double-action hydraulic hoist system. Overall height of telescoped uprights, 83 in.; uprights extended, 145 in.; max lift of forks, 126 in. Overall length with 36-in. forks, 118 in.; width, 42½ in. Hoist speed with full load: going up, 20 fpm; down, 40 fpm. Travel speed with full load, 5 mph. *Elwell-Parker Electric Co., Cleveland, O.*

SHEET FEEDING TABLES: Utilize scissors type arrangement for lifting mechanism. Table elevates from 18 to 36 in., with power supplied to hydraulic rams by foot-operated hydraulic pump or motor-driven pump. Caster-mounted tables available in capacities of 6000 and 10,000 lb. *Lyon-Raymond Corp., Greene, N. Y.*

ELECTRIC FORK TRUCK: Capacity, 2000 lb at 24-in. load center. New Clipper model features increased speed, fingertip directional lever, automatic acceleration, "deadman" safety controls and pivot-mounted steering axle. Specifications: overall length, less forks, 63⅝-in.; width, 34½ in.; turning radius, 61½ in.; minimum intersecting aisles, 57 in.; overall height with forks down, 83 in.; max lift, 124 in. *Clark Equipment Co., Industrial Truck Div., Battle Creek, Mich.*

AUTOMATIC PRESS UNLOADERS: Six metal jaws to accommodate various die and lift conditions encoun-

tered in unloading of stamping presses. Designed for three "Iron Hand" models. Jaw types include: vise type for parts raised 2 or more in. above die surface; chisel type; hook type for use where gripping edge is lacking; Neoprene-tipped gripper; pivot type that swings part laterally to clear tight die areas; and confined type which locks vertical flange stamping without damaging edges. *Sahlin Engineering Co., Birmingham, Mich.*

SHOP CRANE: Permits 5000 lb to be hoisted and moved by one man. Includes hydraulically adjustable center post to vary lifting height. Winch raises load; 7 turns of winch handle raises load 1 ft. Max lift, 9 ft 8 in. Mounted on 8-in. roller bearing wheels. *Allied Manufacturing & Sales Co., Chicago, Ill.*

Power Plant Equipment

DIESEL ENGINES: Line of diesel engines for stationary service. EV series built in 2, 3, 4, 5, 6, and 8-cylinder models developing 45 hp per cylinder at 600 rpm. Supercharged EVS series built in 4, 6 and 8-cylinder sizes developing 60 hp per cylinder at 600 rpm. All engines have bore and stroke dimensions of 8¾ x 11½ in. *Lister-Blackstone Inc., New York, N. Y.*

Processing Equipment

MERCURY CLEANER: Mercury oxifier has capacity of 150 lb, can clean mercury of dissolved base metals, up to 600 lb per day in 150-lb batches. When used in combination with type H filter, unit also eliminates floating impurities such as water, oil, dust and acid residue. Electric motor drives paddles which whip mercury into thick spray for several hours. Base metals oxidize and form powder which precipitates as dust and can be filtered out. Residual contamination less than one part in one million. *Bethlehem Apparatus Co., Hellertown, Pa.*

Testing and Inspection

METALLOGRAPH: Model OPL metallograph for rapid, low-cost structural analysis of metal samples and other substances. Occupies table space 12 by 12 in., stands 18 in. high. Staging table lifts for lens changes; lenses quickly inserted and removed, need no centering. Refocusing not required when lenses are changed. Sample observed through binoculars or on ground glass screen. Photographs taken on 35-mm roll film by swinging image from ground glass to camera. *F. T. Griswold Manufacturing Co., Wayne, Pa.*

BOOKS FOR NEW PRODUCTION IDEAS

PLASTICS IN ENGINEERING By John Delmonte. Here's a brand new "how-to-do-it" book of technical data that will help you determine the RIGHT plastic material, the RIGHT mold design and the RIGHT fabricating technique to solve your plastics problem. PLASTICS IN ENGINEERING puts scores of up-to-the-minute facts about plastics right at your fingertips: How strong various plastics are—for what uses they are best suited—how to design plastic parts—how these remarkable synthetic materials are molded, extruded, laminated, cast—how much heat they will stand—how to guard against failure—how to machine plastics—and other vital factors that are so essential to satisfactory plastics performance. Contains 656 pages . . . 168 illustrations . . . 81 tables. Third edition. *Price $10.*

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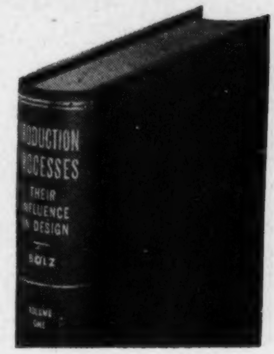
How Much Guesswork Are You Putting into Your Designs?



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